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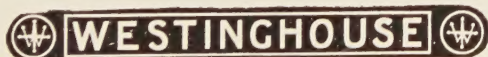
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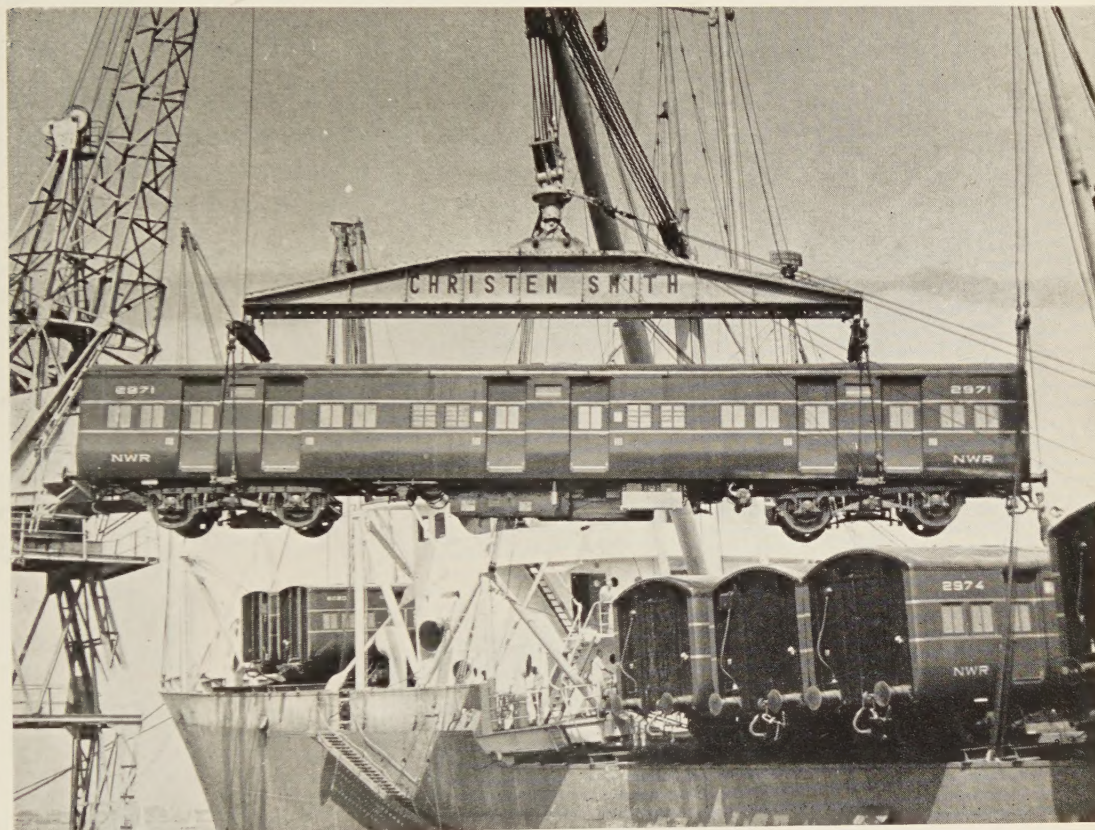
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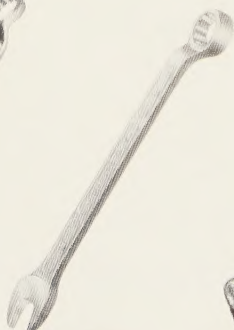
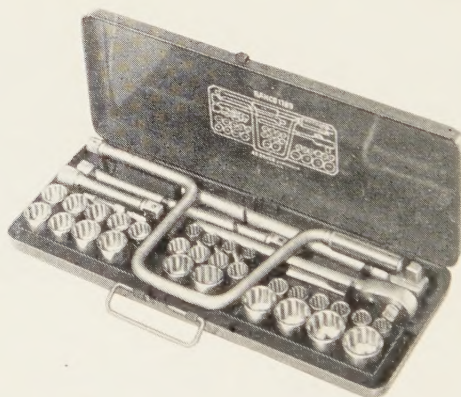
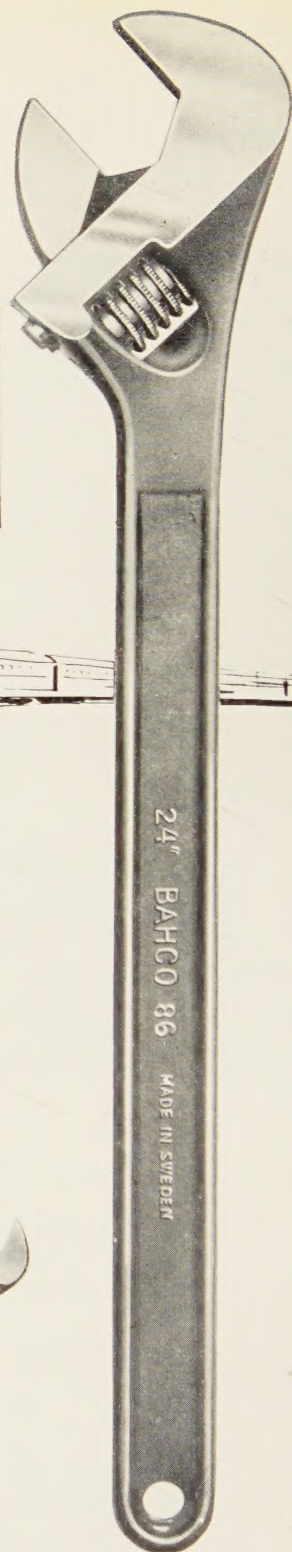
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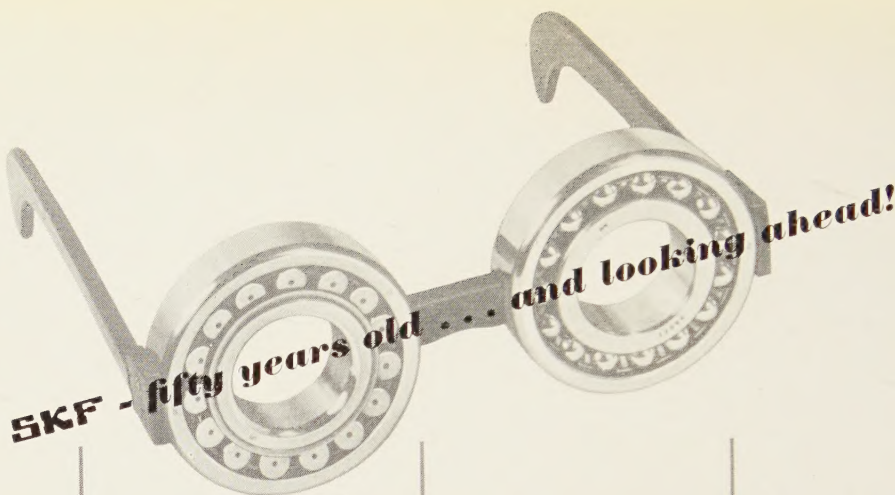
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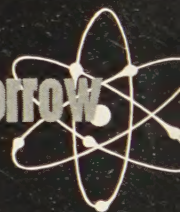
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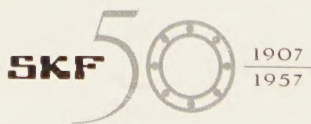


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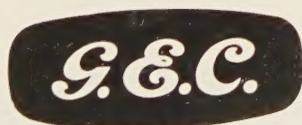


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BULLETIN

OF THE

INTERNATIONAL RAILWAY CONGRESS

ASSOCIATION

(ENGLISH EDITION)

[625 .142 .4 (485)]

Concrete sleeper 101 with Fist rail chair,

by Tord FOGELBERG,

Chief Division Engineer, Swedish State Railways, Nässjö, Sweden.

The Swedish State Railways decided in the autumn of 1950 to investigate the possibilities of changing, at least to a certain extent, from wooden sleepers to concrete sleepers. The rising prices of timber products and the difficulties of adequately supplying the demand for wooden sleepers had contributed to this decision. No great experience in the use of concrete sleepers was available in Sweden. At an earlier time, in 1942, a small number of concrete sleepers had been laid on the main tracks of the line Stockholm-Malmö, in the vicinity of Hässleholm and in the neighbourhood of Malmö. These sleepers were made of bonded wire-prestressed concrete. Their dimensions had been chosen quite simply so as to be adapted to those of the ordinary wooden sleepers, viz., a length of 2.7 m, a width of 200 mm, and a height of 160 mm. The rails were secured to the sleepers by means of spring spikes, which are commonly used on the main tracks of the Swedish State Railways. The spikes were driven into wood plugs, which were embedded in the concrete. These sleepers are still in use on the tracks, and have turned out very well. They do not exhibit any visible signs of

deterioration other than a number of hair cracks, mostly in the middle parts of the sleepers. However, it has proved difficult to ensure that the spikes should remain firmly fixed in the wood plugs, which were liable to become loose during dry weather.

The Royal Swedish Board of Railways entrusted the Author with an investigation of the concrete sleeper situation in Europe in 1951. This investigation consisted of two parts, viz., first, studies of the available literature, and second, visits to Great Britain, France, and Germany, i.e. the countries where the development in this field had advanced farthest. It was found, however, that the opinions on the subject in question were not a little divergent. Nevertheless, all specialists agreed that concrete sleepers can be expected to have a considerably longer useful life than wooden sleepers. Furthermore, they admitted that concrete sleeper tracks possess a greater stability, and hence require a smaller amount of maintenance, than wooden sleeper tracks, because the former are so much heavier, and are provided with stronger rail chairs. Last but not least, the experts agreed that concrete sleepers enable a

change-over to all-welded tracks to be carried out in a better and more reliable manner, so that the rail joints — the weakest points of the track — can be eliminated. All the same, the specialists expressed different opinions on the design and some other features of concrete sleepers as well as on the design of rail chairs.

Original requirements.

The above-mentioned investigations were used as a basis in drawing up the main requirements for concrete sleepers specially adapted to conditions prevailing in Sweden, where the subgrade has a relatively low stability, since it largely consists of clays and moorland soils as well as frost-heaving soils. These requirements were as follows :

1. Concrete sleepers shall be made of prestressed concrete.
2. Concrete sleepers shall be elastic, and shall be deformable to a certain degree.
3. The rail chair shall be doubly elastic, that is to say it shall permit the rail to move upwards as well as downwards.
4. Concrete sleepers shall be superior in respect of electrical insulating properties to wooden sleepers when the latter are at their worst.
5. The total cost of concrete sleepers and rail chairs shall be reasonable.

Concrete sleepers should be designed so as to be free from cracks. Accordingly, the prestressing force should be so great as to prevent crack formation at the maximum bending moments which occur in actual use on tracks. The reason for this requirement is that the absence of cracks guarantees a maximum useful life of sleepers. However, even if cracks should

develop under certain conditions, this will not be so important in a *prestressed* concrete sleeper. In fact, when the load is removed from such a sleeper, the cracks close, so that the water, if any, cannot make way into the interior of the concrete.

A concrete sleeper, like any other concrete structure that is partly embedded in earth, is for easily comprehensible reasons very sensitive to damage caused by water in the event that an open crack occurs in the structure. A crack in an ordinary reinforced concrete sleeper does not close after the removal of load, but remains open. It is readily understood that the useful life of such a sleeper is considerably shortened if water percolates through the crack into the interior of the concrete, and produces a bursting effect as a consequence of corrosion and frost. It is true that a sleeper made of ordinary reinforced concrete can be designed so that it will remain free from cracks, but such sleepers are unnecessarily expensive because they require heavy reinforcement.

Sleepers rest on a bed which is not quite firm, but is liable to displacements caused by subsidence or heaving of the subgrade on account of unfavourable soil conditions, or by freezing, or by precipitation, or, finally, by normal disturbances of the ballast itself under the action of trains. Under Swedish conditions, it is particularly the effect of frost on frost-heaving soils that is a serious handicap in this respect, and that constitutes a problem which must be taken into account in sleeper design. It is important, however, to take preventive measures of various kinds in order that frost-heaving may be wholly or partly obviated. This intricate problem lies beyond the scope of the present article. Suffice it to say

that there are, in the main, three kinds of these preventive measures, which can also be combined :

1. The frost-heaving soil is removed, and is replaced by some other soil which is not frost-heaving, such as gravel.
2. The subgrade is insulated by covering it with a layer of gravel which is sufficiently thick to prevent frost from reaching down to the frost-heaving soil.
3. The ground water table is lowered by means of underdrainage.

If such preventive measures are not taken, and if the frost heave is too great and non-uniform, then it is scarcely worth while to use concrete sleepers because their design will be too difficult. For all that, even if the effects of frost are wholly or partly eliminated, some disturbances in the sleeper bed will be inevitable, and it is therefore necessary that the sleepers should comply with severe requirements in respect of elasticity. Moreover, a certain resilience of the sleepers is desirable because it facilitates track laying; otherwise the levelling of the sleeper bed, i.e. the top ballast, will require work that is excessively accurate, and hence much too expensive.

The operating results obtained on the main tracks of the Swedish State Railways show that elastic rail chairs have great advantages over the conventional rail spikes. The tests and investigations made by the French State Railways (S.N.C.F.) have also confirmed that a doubly elastic rail chair is to be preferred because it maintains the rail, the chair, and the sleeper in permanent contact, and thus prevents the formation of clearances between these parts. This decreases the

wear of the sleeper bed, and hence contributes to a reduction of other operations required for track maintenance. Rigid rail fastenings, such as screw fastenings and spikes, are liable to become loose in a short time. After that, clearances are formed between the rail and the sleeper, which are therefore exposed to increased vibrations. Vibrations cause deterioration of sleepers and disturbances in track position. These questions have been studied, particularly in France, by means of very extensive and time-consuming tests, which were made in the field, on actual tracks, and under idealised laboratory conditions, with the aid of vibrating machines.

The Swedish State Railways are increasing, or will increase, the traffic capacity of their main tracks by installing line block sections and remote control equipment. The rails are used in this connection as conductors for the track circuit currents. If wooden sleepers are to be replaced by concrete sleepers, it is therefore necessary to make sure that this change-over does not involve any drawbacks from a signal engineering point of view, and does not impede the measures taken in order to increase the traffic capacity. Concrete is a better electric conductor than wood. Consequently, special arrangements are required with a view to ensuring adequate insulation. Attempts have been made to increase the electric resistance of concrete by the addition of special agents, but they were not found to be sufficiently effective. In conformity with the tests which have been carried out, the insulation should be fitted in the rail chair, which should therefore be designed so as to comply with the electrical requirements.

The first cost of wooden sleepers in Sweden is relatively low. Accordingly, in spite of the fact that concrete sleepers are far more advantageous than wooden sleepers from an engineering point of view, it is necessary to keep the price of concrete sleepers on a low level if they are to compete successfully with wooden sleepers. However, the designer is much freer to adapt concrete sleepers to the forces which act on tracks than he is in the case of wooden sleepers. Those concrete sleepers which were in more or less general use in the early 1950ies did not quite fulfil the requirements touched upon in the above. In order to collect empirical data on these problems, four test track sections were laid in the spring of 1953 in accordance with a decision of the Royal Swedish Board of Railways. Each of these track sections was 500 m in length. The rails, 50 kg per metre in nominal weight, were welded together so as to form continuous track sections. All concrete sleepers were monolithic and prestressed.

The tests made on these track sections are briefly described below. The characteristics of the test track sections were as follows :

- I. Length 500 m. Sleeper manufacturer: DYWIDAG, Hamburg, Germany. Sleeper type : B 12. Rail fastening : Screw in wood plug.
- II. Length 500 m. Sleeper manufacturer: DYWIDAG, Hamburg, Germany. Sleeper type : B 12. Rail fastening : FIST rail chair.
- III. Length 500 m. Sleeper manufacturer: Betongindustri, Stockholm, Sweden. Sleeper type : Bonded wire-prestressed concrete. Rail fastening : Screw in wood plug.
- IV. Length 500 m. Sleeper manufacturer: A-Betong, Vislanda, Sweden. Sleeper type: « Hairpin-reinforced » concrete (see below). Rail fastening: FIST rail chair.

The test track sections follow one another without break on the line Nässjö-Stockholm, a part of the main line Malmö-Stockholm, where the maximum train speed is 130 km per hour. Figure 1 shows a portion of the test track at the junction of the sections II and III. (The section III is nearest to the spectator.)

Each sleeper in the test sections I and II is reinforced with two bars, 18.6 mm in diameter, which are subjected to a total prestressing force of 24 metric tons. Each sleeper in the test section III is provided with wire reinforcement, 2 mm in diameter, and the total prestressing force is also 24 metric tons. The « hairpin » reinforcement of each sleeper in the test section IV consists of a butt-welded single steel bar loop, 19 mm in diameter, and is prestressed by means of wedges, which produce a force of 26 metric tons.

Experimental results.

Continuous measurements of the vertical, longitudinal, and lateral displacements of the track in the various test sections have been made in the course of different months. The temperature of the rails has varied from -30°C to $+50^{\circ}\text{C}$ during the period of observation.

A number of small cracks in some sleepers were detected some time after laying of the test track sections. These cracks, which were submitted to running observation, have not become larger, and the sleepers are still in good condition. It seems that any appreciable reduction in the useful life of these

sleepers is not to be apprehended since the concrete is prestressed. A rail fracture occurred in the test section II at a temperature of about -25°C during the first winter. In spite of the low temperature, the width of the fracture

and IV, counteracted rail creep better than the screw fastening, which was employed in the sections I and III. The movements in the other directions, i.e. lateral and vertical, were negligible with the exception of those in a part of the



Fig. 1.

opening did not exceed 8 mm. This event has shown in a palpable manner that the rail chair can keep the rail under control so as to prevent it from contraction.

Furthermore, the measurements of the longitudinal movements of the rails demonstrated that the « Fist » rail chair, which was used in the test sections II

and IV, counteracted rail creep better than the screw fastening, which was employed in the sections I and III. It is to be noted, however, that the subgrade in this section is not very satisfactory, and the greater need of adjustment is consequently due to this circumstance.

In order to study the electric resistivity of the sleepers, repeated measurements

have been made in the different test sections during one year. However, the results of these measurements were not good enough to enable the signal engineers to approve the sleepers so far as insulation was concerned. The comparison track section, which was equipped with ordinary wooden sleepers, exhibited a resistance which was higher throughout than that of the test sections. In view of the low and unsatisfactory electric resistance observed in the test sections, it was therefore necessary to investigate the possibilities of improving the properties of the concrete sleepers in this respect. Now the Fist rail chair was considered to be superior to the screw fastening. Moreover, as has already been mentioned, it was deemed most appropriate to fit the insulation in combination with the rail chair. Accordingly, special attention was concentrated on the solution of the insulation problem in connection with the Fist rail chair. This rail chair, which is shown in figure 4, will be described at some length further on in the present article.

The Fist rail chair affords fairly great possibilities of choosing an appropriate place for the insulation. The rails in the test track sections equipped with Fist rail chairs were insulated provisionally by means of ordinary insulating tape, which was simply wrapped round both ends of the rail chair pin at its points of contact with the stirrup. However, it is preferable either to fix the insulation on the exterior surface of the pressure-equalising steel tube, which is embedded in the concrete, or to insulate the pin itself.

Two additional test track sections, 1 km in length each, provided with con-

tinuously welded rails, concrete sleepers, and Fist rail chairs were laid in the autumn of 1954. The main purpose of these test sections was to elucidate the insulation problem. The top ballast consisted of crushed stone in one of these sections (see fig. 2) and of gravel in the other so that the effect of the ballast might be investigated at the same time. The exterior surface of the pressure-equalising tube was thoroughly cleaned, and was then covered with a double coating of insulating lacquer (isocyanate lacquer), which was baked at a temperature of 120° C.

In order to cut down the cost of the sleepers themselves, the prestressing force applied to these sleepers was reduced to 15.6 metric tons. At the same time, the height of the sleeper was slightly increased. The reinforcement consisted of a single bar, 17.7 mm in diameter, and the yield point of the steel was 90 kg per mm². The results obtained in these test track sections are on the whole good. However, the number of sleepers with cracks in their middle parts was found to be somewhat greater in these test sections than it had been in those which were laid in 1953. Some cracks were so dangerous that it was considered necessary to replace the sleepers. On the other hand, no cracks have been observed in the concrete blocks themselves. Nor has the track position undergone any changes.

From an electrical point of view, the observed values were so good that they were superior to those obtained in the comparison track sections, which were provided with wooden sleepers. No appreciable difference in electric resistance between the sleepers laid in crushed stone

and those resting in gravel was to be demonstrated.

For all that, the electricians of the Swedish State Railways desired that the sleepers should furthermore be tested for puncture strength under the action of the high voltages which can sporadically

machine, which was similar in principle to that introduced by the S.N.C.F. French investigations have shown that this machine produces stresses of the same magnitude as those which occur on tracks. It was found in these tests that a few hours of exposure to vibrations in



Fig. 2.

occur in electric railway operation. These tests, which were made in the laboratory, showed that the insulating lacquer did not comply with the stipulated requirements under certain conditions. Furthermore, the question of long-time fatigue tests was also taken up in this connection. These tests were carried out in a vibrating

the machine were already sufficient to cause cracks in the middle part of the sleeper. These cracks spread little by little, and finally united with other cracks, so that the middle part of the sleeper was split into fragments, which became loose. As a result of this, the prestress was removed, and the sleeper failed. The

vibrating machine operated at a frequency of 3 000 cycles per minute, and developed a force of 4 metric tons per rail bearing. Thus, the middle part of the sleeper proved unable to withstand the vibrations generated by the testing machine. It was therefore considered necessary to change the design of the sleeper, and at the same time to revise the assumptions on which this design was founded.

New requirements.

On the basis of the results obtained in the test track sections as well as in the other tests, and in consideration of all those problems which had emerged in this connection, the Author altered the requirements which had been stipulated previously. The new main requirements were formulated as follows :

1. Concrete sleepers shall be made of prestressed concrete so as to prevent crack formation, and thus to guarantee a maximum useful life.

2. The middle part of the sleeper shall combine elasticity with great strength, and should therefore be wholly or partly made of a material which has a higher modulus of elasticity than concrete, e.g. steel.

3. The rail chair shall be doubly elastic, that is to say, it shall permit the rail to move upwards as well as downwards. Furthermore, it shall also possess a certain elasticity in a lateral direction. At the same time, it shall completely prevent the rail from longitudinal creep. It shall exert a sufficiently great force to press the rail on the sleeper. The continuity of this force shall be ensured

in such a manner that no supervision and maintenance will be required.

4. The electrical insulation resistance of concrete sleepers shall be as high as possible, and shall be superior to that of wooden sleepers. It shall be ensured by such means as to exclude the possibility of insulation failure, e.g. due to fatigue, flashover from the electric contact line, incorrect fitting, or displacement of the components entering into the rail chair. If any faults should occur in the electrical insulation, then it shall be possible to remedy them in a simple way, without having to discard the sleeper.

5. The total cost of concrete sleepers and other accessories shall be reasonable in spite of the fact that experience has shown concrete sleepers to be superior to wooden sleepers in certain technical respects.

Concrete sleeper 101.

After very extensive tests, these new requirements resulted in a new type of sleeper — the concrete sleeper 101 with the Fist rail chair, which is briefly described in what follows.

As is seen from figure 3, the sleeper 101 consists of two concrete parts, which are joined together by means of a concrete-filled steel tube, 63 mm in outside diameter and 3.5 mm in wall thickness. The sleeper is reinforced with a prestressed steel bar, which passes through the steel tube, and is anchored at the outer ends of the concrete parts by means of washers and nuts. This method of reinforcement has formerly been used for the monolithic full-length sleeper, and is similar in principle to the Karig or the Franki-Bagon system. The

reinforcing bar in the sleeper is subjected to a prestressing force of 13.5 metric tons, which thus acts on the whole sleeper. The bar material can be chosen from among several steel grades. Use has so far been made of two steel grades, viz., first, rolled and subsequently cold-drawn steel bars, 17 mm in diameter, and second, silicomanganese steel bars, 19 mm in diameter, which were heat-treated after rolling in order to relieve stresses, if any. It was considered that this heat

to any appreciable corrosion. French experiences have also shown that the tie bar of the R.S. sleeper, which consists of a structural steel shape, gives good results in this respect. Furthermore, a tube has the most favourable cross-sectional area so far as corrosion is concerned. Moreover, the tube is submitted to continuous pressure, and this circumstance contributes to prevention of corrosion. However, if protection from corrosion is desired under certain

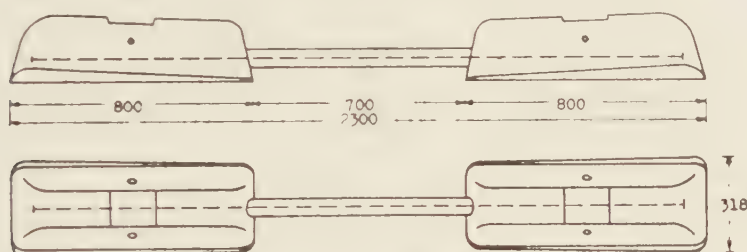


Fig. 3.

treatment would reduce the risk of sudden failure.

The choice of the steel grade for the tube is dependent on the requisite elasticity of the tie constituting the middle part of the sleeper. Cold-drawn and tempered tubes having a yield point of at least 44 kg per mm² can be recommended for use under Swedish conditions. Since the tube is exposed to the weather, and is placed directly above the ballast, it might be supposed that the tube is laid open to the risk of corrosion. Experience has shown, however, that corrosion is not a factor of vital importance in areas which are not situated in coastal regions or in the neighbourhood of certain industrial plants. Rails subjected to the action of traffic are not liable

conditions which involve a great danger of corrosion, then it is advisable to galvanise the tubes. This can be done at a low cost in connection with the tempering process. The sleeper shall be made of high-grade concrete having a compressive strength of at least 600 kg per cm² at 28 days and a modulus of rupture of at least 65 kg per cm² at 7 days.

The compressive force is transmitted between the ends of the tube to the tube itself as well as to the concrete contained in the tube. Thorough investigations and tests, which were made by means of strain gauges attached to the exterior surface of the tube in order to measure the stresses, have shown that the stress in the tube gradually increases as the prestress of the reinforcing bar becomes

greater. Then the stress in the tube decreases by an amount which is comprised within the limits from 5 to 10 per cent of the maximum value. The greater part of this decrease occurs as early as during the first two or three days after prestressing. This reduction in stress may probably be attributed to two causes, viz., first, the fact that the load is partly transferred from the tube to the concrete in its interior, and second, the circumstance that the prestress in the reinforcing bar slightly decreases after the prestressing has been completed.

On the other hand, no appreciable reduction in stress occurs after the sleeper has been post-stressed. Measurements have shown that the stress in the tube may be expected to vary within the range from 9 to 15 kg per mm². This apparently great dispersion is largely caused by the tolerance involved in the operation of the tensioning tool which was used for prestressing.

The compressive force acting on the concrete in the interior of the tube gives rise to a stress in an outward radial direction. This stress increases the rigidity of the tube as well as its resistance to plain and lateral buckling.

The ends of the tube have been expanded so as to form a cross section which is very nearly square. This ensures complete torsional rigidity of the restraint. In order to secure a still closer structural interaction between the concrete and the ends of the tube, the concrete at the tube ends has been reinforced with a spiral made of a steel bar, 5 mm in diameter. Since the middle part of the sleeper is prestressed, the elastic properties of the sleeper are exceedingly good, with the

result that the concrete blocks can adapt themselves to alterations of the ballast bed. As has previously been mentioned, the ballast is deformed indirectly, owing to the displacements of the subgrade, and directly, on account of the fact that the ballast is not homogeneous in itself. The rolling load, which is transmitted through the rails and the sleepers, causes changes in various points of the ballast under the sleeper. The sleeper must be able to adapt itself to the normal displacements of the ballast without undergoing any permanent deformations. The sleeper 101 can be compared to a powerful buffer spring, which remains unchanged at rest, but becomes resilient under the action of load.

Fist rail chair.

The requirement stated under point 3 in the above has previously for the most part been fulfilled by the use of the Fist rail chair. However, this requirement is closely bound up with the design of the sleeper itself. In other words, the design of the rail chair must be taken into account in designing the sleeper itself. In addition to the principal condition stated under point 3 in the chapter « New Requirements », the rail chair must also satisfy the following secondary requirements :

a) For natural reasons, the rail chair shall not comprise any wooden components. Such components may be expected to have a shorter useful life than the other parts of the rail chair, and the replacement of these components will therefore involve high costs. On account of variations in its moisture content, wood is liable to expansion by

swelling, which involves the risk of stresses in the concrete, or to contraction by drying, which involves the risk of loosening of the rail chair itself.

b) The rail chair shall grip the rail so firmly that it effectively prevents rail creep, if any, without necessitating the

that is as small as possible. Furthermore, the rail chair shall be easy to fit and to remove.

d) The rail chair should preferably not submit the sleeper to any moments which are added to the moments caused by the wheel loads.

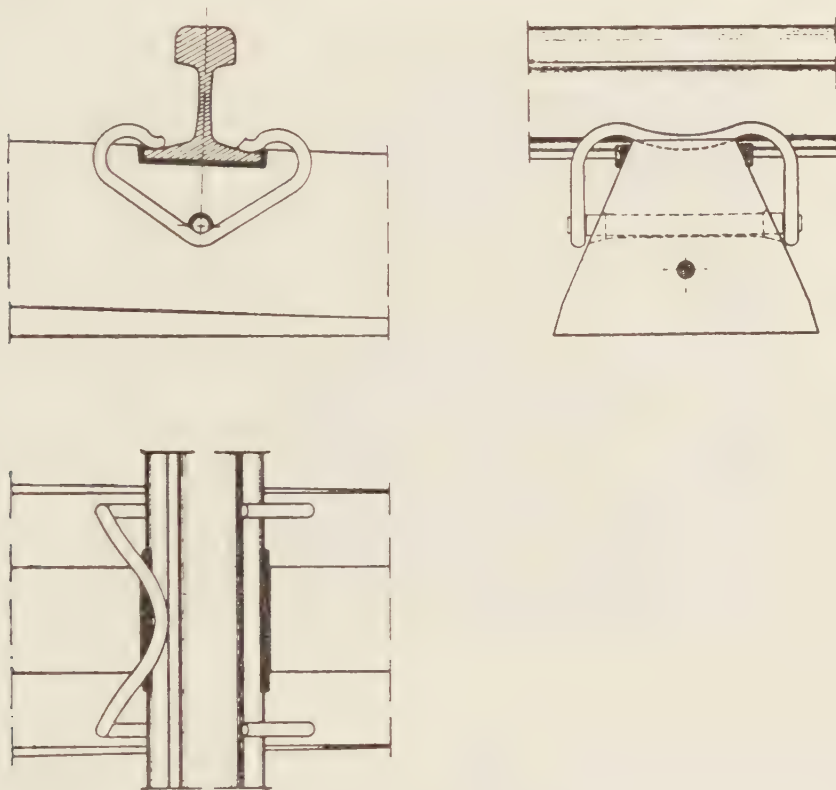


Fig. 4.

use of any special devices for rail creep prevention. If this requirement is fulfilled, then this facilitates a change-over to continuous rails, i.e. to welding of all rail joints.

c) The rail chair shall be simple, and shall comprise a number of components

e) It is not desirable to provide the sleeper with vertical holes for attaching the rail chair, since these holes would weaken the sleeper at the points where the maximum moments are applied. Furthermore, moisture can easily make way into such holes, and this involves

the risk of deterioration due to frost and weathering. In other words, the top surface of the sleeper should be smooth.

As is seen from figures 4 and 5, the Fist rail chair consists of a stirrup, which is made of a round spring steel bar. 15 mm in diameter. The stirrup is bent so that it exerts pressure on the rail at

and reduces the strength of the concrete block in a very small degree. Furthermore, the hole is reinforced with a steel tube, which is tapered at the ends. This tube also serves to distribute the pressure exerted by the pin. Each end of the pin is indented so as to form a notch, which prevents the pin from moving on account



Fig. 5.

three points of contact, and so that it produces a force of at least 1 500 kg pressing the rail down against the sleeper. The stirrup is fitted on the sleeper by means of a pin, which is also made of spring steel. The pin, in its turn, is inserted in a horizontal hole in the neighbourhood of the neutral layer of the concrete block. The hole is situated under the rail foot or flange so as to be protected from the action of the weather.

of the vibrations due to trains. Thus, the rail chair does not comprise a single wooden component.

The clamping force produced by the rail chair is centrally applied under the rail foot. This force gives rise to moments and stresses which are opposed in direction and in effect to the action of the wheel loads on the rails. Consequently, this circumstance increases the load-carrying capacity of the concrete block.

The rail is supported on a fluted rubber pad, which is fitted in a slot in the top surface of the sleeper. The rubber pad is provided with bent-up and curved edges, which clasp over the side edges of the rail foot. The rubber pad acts as a shock absorber, which greatly damps the vibrations transmitted to the sleeper.

The rail chair does not comprise any bolts, and is held in place by its own compressive force, which acts directly on the rail foot. Moreover, the rail chair rests in a stable position on account of its shape, and cannot become loose. Accordingly, the rail chair does not require any maintenance or supervision. The magnitude and the permanence of the clamping force are decisive factors in connection with change-over to continuously welded rails, which represents a marked trend in modern railway construction. Since the compressive force is great, and since the various components of the rail chair are resilient, the rail chair is elastic in all directions, and hence also in a lateral direction, but this elasticity does not lead to any changes in the position of the rail on the sleeper. The rail chair comprises only two separate parts, and is so designed and constructed as to preclude the possibility of any mistakes in fitting. The rail chair is easily fitted with the help of a hydraulic tool which has been specially designed for this purpose.

Electric resistance.

The requirements in respect of adequate electric safety, including sufficiently high resistivity, stipulated by the signal engineers have in some measure delayed the solution of the whole concrete

sleeper problem. As has previously been mentioned, the isocyanate insulation was not approved because it was found that even relatively low voltages were already sufficient to cause puncture. An accurate analysis of the insulation problem has shown that the insulation to be fitted on the Fist rail chair should comply with the following requirements :

1. High electric resistance both on dry and on wet tracks.
2. Adequate electric puncture strength both on dry and on wet tracks.
3. Adequate mechanical strength under the action of both static and dynamic stresses.
4. Adequate resistance to light action.
5. Adequate resistance to attack by chemical and biological agents.
6. Adequate resistance to fluctuations in temperature.
7. Adequate resistance to ageing.

Furthermore, it is desirable that the insulation should be easy to replace in the event of faults.

The requirement that the electric resistance should be high is self-evident in view of the fact that the sleepers are closely spaced, and therefore form a large number of parallel paths for the electric current. This requirement not only concerns the insulation of the rail chair itself, but is also applicable in an equally high degree to the risks involved in the fact that the insulation clearance can be bridged over, e.g. by accumulation of particles produced by wear of brakes. Adequate puncture strength is required in consideration of the circumstance that the rail voltage to earth on electric rail-

ways can under certain unfavourable conditions be as high as 1 500 V. It is to be noted, however, that this figure represents a maximum value which is met with in very unfavourable cases only. Adequate mechanical strength is necessary seeing that the rail chair exerts a pressure ranging from 1 500 to 2 000 kg. Moreover, the stresses due to vibration are to be added to this pressure.

Most insulating materials are sensitive to light, particularly to ultraviolet rays. On the other hand, the insulation will scarcely be exposed to solar radiation. Attention must furthermore be paid to attack by chemical agents, e.g. uric acid, humic acids, leakage from railway cars, and the like. Attack by biological agents is also imaginable. The temperature in Sweden varies within the limits from -40°C to $+50^{\circ}\text{C}$, so that the insulating material has to comply with severe requirements in this respect. Finally, the material must also offer a great resistance to ageing, which is dependent on various factors.

A very great number of materials which might be used for this purpose have been tested both on tracks and in laboratories. As a result of these tests, the most appropriate and best solution was considered to be the use of a fabric-base laminated phenolic insulation, 2.5 mm in thickness, which was pressed on the pin itself. As has already been pointed out, the rubber pad has been provided with bent-up edges which clasp over the top surface of the rail foot so as to increase the insulation clearance between the sleeper and the rail. Measurements made on tracks have shown that the electric resistance decreases to a very high degree

as the moisture content becomes greater. Furthermore, the electric resistance is also influenced by the temperature so that the resistance reaches a minimum immediately above the freezing point.

The curves shown in the graph in figure 6 represent the observed electric resistance, expressed in ohms per kilometre of track length, at a maximum moisture content and at a voltage of 6 V. One of these curves relates to a wooden sleeper track consisting of rails, 50 kg per metre in nominal weight, fastened with spring spikes and washers, and laid in crushed stone ballast. The other curve refers to a concrete sleeper track consisting of rails, 50 kg per metre in nominal weight, equipped with Fist rail chairs, which are provided with phenolic-insulated pins and rubber pads. As is seen from this graph, the resistance of the concrete sleeper track reaches a minimum point at a temperature of about 0°C . This implies that the track is covered with melting snow up to the rail foot. The resistance minimum point of the concrete sleeper track is situated considerably higher than the curve for the wooden sleeper track.

Moreover, the phenolic insulation not only fulfilled the requirement concerning the puncture strength, but also proved to afford a wide safety margin in this respect. In order to test the resistance to ageing, the whole unit assembly, i.e. the concrete sleeper 101 and the Fist rail chair with a phenolic-insulated pin and a rubber pad, was submitted to vibration in the above-mentioned vibrating machine during about 700 hours, see figure 7. In this test, the machine was operated at a frequency of 1 000 cycles per minute.

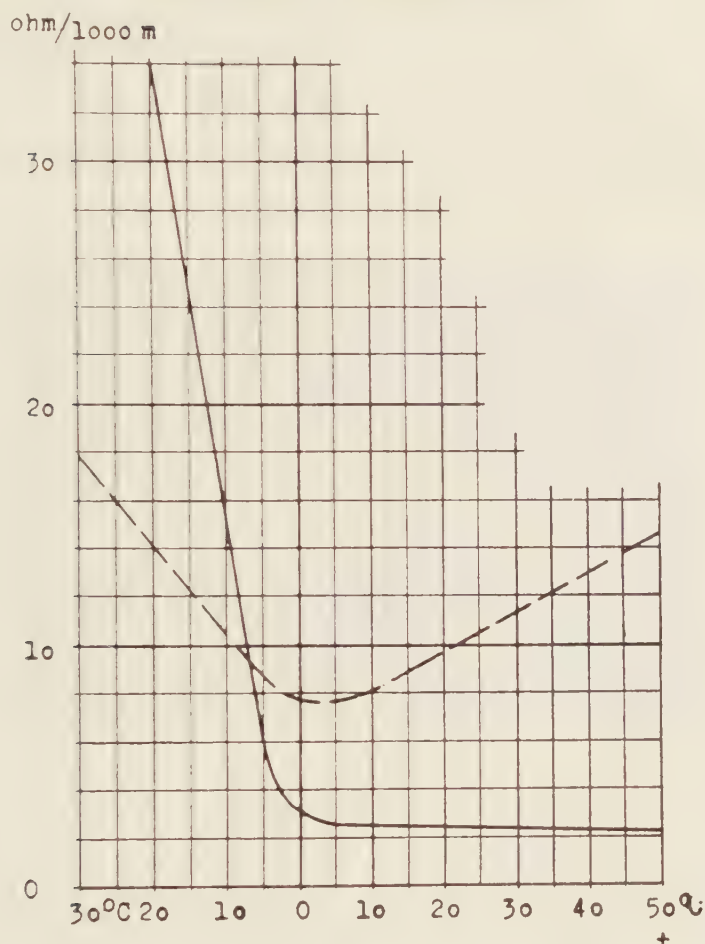
Elektriska ledningsmotståndet.

Fig. 6.

Electrical resistivity.

Ohms per 1 000 metres

30° C ... 50° C

This graph represents a comparison between

— creosote-impregnated wooden sleepers with rail spikes

and

— — — concrete sleepers with Fist rail chairs and pin insulation
at a maximum moisture content and a voltage of 6 V.

The temperature has varied from -30°C to $+20^{\circ}\text{C}$ during this period. It is considered that this test corresponds to

the tresses met with on a track handling very heavy traffic (10 000 000 gross metric tons per year) in the course of 50 years.

After this test, no cracks whatever and no other traces of deterioration were to be observed on the sleeper, the rail chair, or the insulation. After that, the sleeper was tested in the laboratory in order to find out whether the electric resistance had undergone any alteration. No change in resistance was to be detected.

the prestressed concrete blocks and the elastic middle part. The fact that this sleeper is able to adapt itself to the irregularities of the subgrade and the displacements of the ballast is very important, particularly on railway lines constructed on an uneven subgrade. Furthermore, the Fist rail chair has the

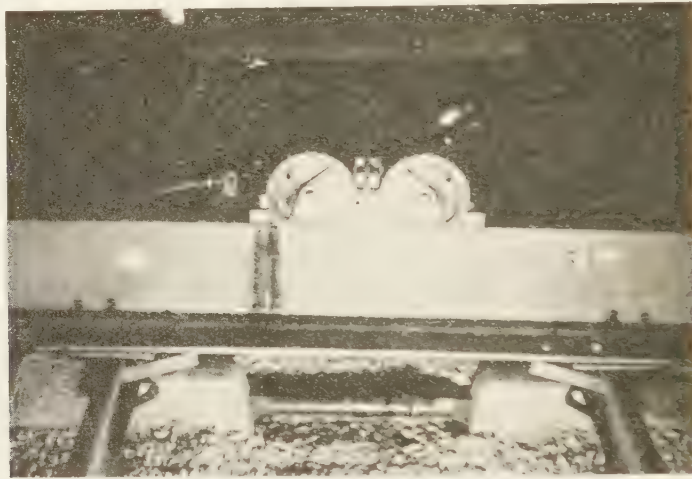


Fig. 7.

The sleeper 101, which may be described as the tie tube concrete sleeper, embodies a new solution of the concrete sleeper problem. This design has obviated the inherent drawbacks of two types of concrete sleepers in common use, viz., the Karig sleeper and the R.S. sleeper. These drawbacks are, respectively, the concrete middle part, which involves the risk of crack formation, and the non-prestressed concrete blocks, which are exposed to the danger of permanent crack development. At the same time, the design of the sleeper 101 combines the respective advantages of the above-mentioned two types of sleepers, viz.,

same good properties as the French R.N. rail chair, which is elastic in an upward direction as well as in a downward direction, which is so strong that no clearance can be formed between the rail and the sleeper, and which effectively absorbs the vibrations due to trains, so that it reduces both track and rolling stock maintenance. Moreover, owing to the absence of bolts, the design of the Fist rail chair has cut down supervision and maintenance of the rail chair itself to a minimum, has ensured a certain elasticity of the rail chair in a lateral direction, and, last but not least, has rendered possible a simple solution of

the insulation problem, which had been a highly crucial question. The combination of the concrete sleeper, the Fist rail chair, and the pin insulation has been tried out and developed in close collaboration with several specialists employed by the Swedish State Railways and two Swedish firms, viz., A-Betong, Vislanda, and Lesjöfors Aktiebolag, Lesjöfors.

It is a well-known fact that engineering never stands still, but continuously advances through development. Accordingly, the design described in the present article is not to be regarded as an ultimate solution of the sleeper problem. Nevertheless, it may be stated that this combination has fulfilled the predetermined requirements in a very satisfactory manner.

Air and gas-filled springs and their contribution to improved vehicle suspension,

by Dr.-Ing. G. A. GAEBLER, Frankfurt-am-Main.

(Eisenbahntechnische Rundschau, September 1956.)

With speed in traffic still increasing and transport carriers endeavouring to improve the comfort of travel, good spring arrangements in road and rail vehicles are of great importance. The latest development in this field is concerned with spring elements having air or gas cushions. Referring to the latest publication on this subject, the author gives a description of the theoretical and practical fundamentals and tells about the first results obtained with air springing on the German Federal Railways.

I. Preliminary notes on the nature and types of springs.

(a) Steel springs.

The spring is a component of mechanical construction which, like the wheel, is quite inseparable from the general construction technique of vehicles. The methods of spring construction and arrangement are already notable by the type of elastic materials used. The classical form, a steel spring, has so far kept its place because of its constructional qualities and favourable operation, i.e. small dimensions, simple design, almost complete absence of maintenance and reliability in service, even under difficult conditions, and it is used in three basic forms: laminated, helical or volute, and torsion bar. The two latter forms do, it is true, generally call for dampers to reduce oscillation because they possess little inherent damping.

Against the known advantages of metal springs, there are some disadvantages, for example :

1. In given constructional dimensions, the steel spring can only be used within narrow limits for different loads and conditions according to its capacity;

2. In certain undamped forms (for example, volute springs) there is a marked

tendency to high-frequency (harmonic) self-oscillation, the cause of which is in the considerable mass of the spring itself. These harmonics can be troublesome because they are the excitation frequencies of the part suspended (for example, the passenger compartment of a coach);

3. Mass inertia of the spring, resulting from the appreciable weight (its own mass), which entails lack of response to very short and very rapid impulses;

4. Ability, resulting from the inert mass, to transmit high-frequency oscillations as well as noise due to vibration of solid masses.

Effort must be made to avoid or at least reduce these disadvantages of the metal spring by using other elastic agents; in this connection particular use is made of *rubber springs* which have recently been improved, as well as with *gas-filled springs* of widely different types and functioning.

(b) Rubber springs.

Rubber-based springs (1) already have a long history. The valuable properties of rubber as a damper against the transmission

(1) Dr.-Ing. E.F. GÖBEL : Berechnung und Gestaltung von Gummifedern (Calculation and design of rubber springs). Springer, publisher, 1955.

of vibration, shock and noise have long been known and used for purposes of springing. Recent work on the chemical composition of types of rubber has resulted in considerably improving their resistance to heat and cold as well as making them less vulnerable to other materials (oil and grease, for example) and in maintaining these qualities within the desired range. Ageing, which takes place through the action of light, rain, ozone, as well as heat, in the form of disintegration, can also be restricted because the constant dynamic stressing of rubber used for springing, so long as it remains within the limits of endurance, opposes it by permanent solidification. Thus, rubber has become an important material for really resilient work.

It has the advantage that a spring of determined constructional dimensions can be used within certain limits for varying loads and stresses when the method of fitting (for example, the angle - which determines the proportions of shear and compression in a spring working under both shear and compression) is carefully chosen. A drawback of this material is, it is true, that a batch of rubber springs working in a determined direction has, because of its size and mass, an elastic effect equal to the axis of displacement perpendicular to the direction of work; this fact is troublesome in certain applications and where it is desirable, entails certain variations of the spring characteristics in the main direction of work, according to the transverse deviation and consequently often not desirable. In addition, the volume of rubber springs is often considerable and consequently, their mass also; as with metal springs, this may result in the transmission of high-frequency oscillations particularly with high specific stresses (kg/cm^2).

Nevertheless, this type of spring has been introduced more and more into general mechanical construction, as well as into the construction of machine tools, and also to a large extent into the construction of road and rail vehicles, where it is used not only to damp out vibration and noise but, with success, as a true spring. The question of

rubber springs for railway rolling stock, has already been studied in the reports of the International Railway Congress Association (2) and in numerous technical publications (3).

(c) *Air and gas-filled springs.*

Air and gas springs are the latest of modern types of spring, if we ignore the use of pneumatic tyres, which have allowed motorised road vehicles to give performances in running and transport which are accepted nowadays as quite normal. However, a comparison of running between vehicles with solid tyres and with pneumatic tyres — or in other words *pneumatic springs* — is sufficient, even to the layman, to show the considerable difference in nature and effect of the two methods of suspension.

First proposals for using air as the elastic component in fact date from the beginning of the century; in a German patent (4) for pneumatic springs, dated 1900, it was

(2) *International Railway Congress Bulletin*, No. 3 of 1955, p. 228-230; No. 9, p. 703-706.

(3) R. JÖRN : Gummifederung für Ackerwagen (Rubber springs for agricultural vehicles). *Kautschuk-Anwendung*, Rubber Stichting, Delft, Holland, 3rd year, No. 3, June 1953. Was brachte uns die Technische Messe Hannover 1952? (What has the 1952 Hannover Fair shown us?). *Verkehr und Technik*, 5th year, No. 6, June 1952, p. 164.

H. WAGNER : Die Verringerung des betrieblichen Erhaltungsaufwandes als Konstruktionsaufgabe (Lower maintenance costs in relation to design). *Stahl und Eisen*, 75th year, 1955, No. 2, p. 85-89.

V. LÄPPL und H. WAGNER : Die Umformerlok für den Westdeutschen Braunkohlenbergbau für Einphasenwechselstrom 50 Hz 6 kV (Converter locomotive for West German lignite mines, 50 cycle, 6 kV single phase). *Elektrische Bahnen*, 26th year, 1955, No. 2, p. 14-17.

Special number 3, E.T.R. : Kautschuk im Eisenbahnbau (Rubber in railway construction). Articles by O. TASCHINGER, A. MIELICH, F. HEGENBARTH, H. WAGNER, J. BERG and R. JÖRN. Editions Carl Röhrig, 1954.

(4) German patent No. 140 111, Class 47a, 3.4.1900 (delivered on 11.4.1903) « Pneumatische Feder » (Pneumatic spring).

proposed to insert in a cylindrical body provided with a piston, a rubber block containing numerous cells filled with compressed air.

II. Theoretical considerations on pneumatic springs.

(a) Theory of spring characteristic.

To make the essential elements of the reasoning we shall follow more easily understood, we shall first give some simple and general notes on the *characteristic curves of springs*.

The equation of a spring characteristic curve is generally written as :

$$P = Fu(s) \quad (1)$$

and characterises the force applied to a spring P as a function of the displacement. On the basis of the relationship between force applied and displacement, we get the expression of the constant of the spring :

$$c = dP/ds \text{ (kg/m)} \quad (2)$$

and the capacity A of a spring :

$$A = \int P \cdot ds \text{ (kg/m)} \quad (3)$$

In the spring diagram in figure 1, the deflections of a spring are shown as abscissae and the force applied as ordinates. The spring may, as is generally the case with simple steel springs, have a straight line characteristic, but it may also be progressive or regressive; a regressive and at the same time decreasing characteristic would give an unstable spring. The *field of static work* in which a spring can be used is generally determined by a minimum load (1) and a maximum load (2). The *field of dynamic work* overlaps the static one according to the displacement of the spring chosen in each case, for example, downwards from the lower load limit (1) to (3), and upwards from the higher limit (2) to (4). The area

between the spring characteristic, the limits of displacement of the spring given by (3) and (4), as well as the axis of the abscissae, measure the power of the spring.

For a gas-filled spring, the fundamental expressions are similar. Firstly, along with equation (2) we keep the preceding form :

$$p \cdot V^{\gamma} = \text{constant} \quad (4)$$

for the pressure (p) (kg/m^2) and the volume (m^3).

Considering now a column of gas which can be shown as in figure 2, by a hollow cylinder filled with gas, on which oscillates

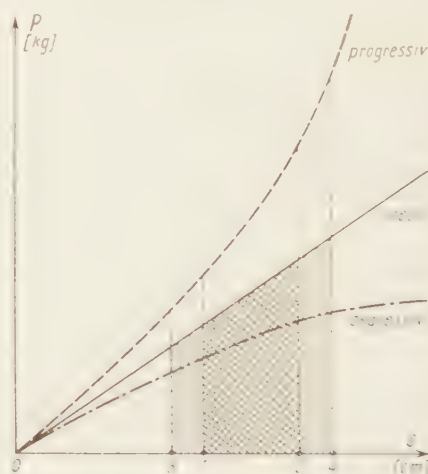


Fig. 1. — Spring characteristics — principles.

a mass m of area F , so that at rest the forces are in equilibrium, the layer of gas of height h_0 corresponding to a volume V_0 and an internal pressure p_0 , carries the weight of the mass m and that when the load is increased by the addition of a impulse force, the mass of gas V_0 is compressed by the amount Δs . This gives the following expressions :

The differential equation of a harmonic undamped oscillation is generally :

$$mx'' + cx = 0 \quad (5)$$

Its solution (5) is expressed (making $x = s$) for a wavelength T :

$$T = 2 \pi \sqrt{\frac{m}{c}} \quad (6)$$

from which is found the frequency, by using the inversion :

$$f = \frac{1}{T} = \frac{1}{2 \pi} \sqrt{\frac{c}{m}} \quad (7)$$

We have :

$$m = \frac{G}{g} \left[\frac{\text{kg} \cdot \text{sec}^2}{m} \right] = \text{mass of suspended load;}$$

c = constant of spring ($\text{kg} \cdot m$);

P = force G kg ;

p = pressure (kg/cm^2);

$V = F \cdot h = \text{volume } (m^3)$;

$$\kappa = \frac{c_p}{c_v}$$

specific heat at constant pressure
specific heat at constant volume

In general, we have for the mass displaced by an amount Δs on the spring made up of a cylinder filled with compressed air (fig. 2) :

$$p_o \cdot V_o^\kappa = \text{constant} = \frac{P_o}{F} [(h_o - \Delta s) \cdot F]^\kappa$$

or

$$P_o = \frac{p_o \cdot V_o^\kappa \cdot F}{[(h_o - \Delta s) \cdot F]^\kappa} \quad (8)$$

The variation of the force P as a function of the displacement s , or the constant of the spring c , then becomes :

$$\frac{dP}{ds} = c = \frac{p_o \cdot V_o^\kappa \cdot F^{1-\kappa} \cdot \kappa}{(h_o - \Delta s)^{\kappa+1}} \quad (9)$$

Relating this value to equation (7) gives the frequency f :

$$f = \frac{1}{2 \pi} \sqrt{\frac{p_o \cdot V_o^\kappa \cdot F^{1-\kappa} \cdot \kappa}{(h_o - \Delta s)^{\kappa+1} \cdot m}} \quad (10)$$

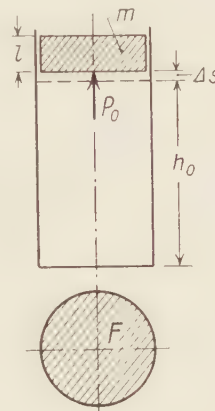


Fig. 2. — Sketch showing basic theory.

If, as is the case in practice, the displacement of the spring Δs is low in relation to h_o and consequently negligible, and if we replace h_o by $\frac{V_o}{F}$, we can write, more simply :

$$f = \frac{1}{2 \pi} \sqrt{\frac{p_o \cdot V_o^\kappa \cdot F^{1-\kappa} \cdot F^{1+\kappa} \cdot \kappa}{V_o^{\kappa+1} \cdot m}} = \frac{1}{2 \pi} \sqrt{\frac{p_o \cdot F^2 \cdot \kappa}{V_o \cdot m}} \quad (11)$$

Taking $p_o = \frac{P_o}{F}$, it becomes :

$$f = \frac{1}{2 \pi} \sqrt{\frac{P_o \cdot F \cdot \kappa}{V_o \cdot m}} \quad (12)$$

or where P corresponds to the weight of the mass m , in other words $P_o = m \cdot g$,

$$f = \frac{1}{2 \pi} \sqrt{\frac{F \cdot \kappa \cdot g}{V_o}} \left[\frac{1}{\text{sec.}} \right] \quad (13)$$

(5) DEN HARTOG : *Mechanische Schwingungen* (Mechanical oscillations). Springer, publisher, 1952, p. 39.

It will thus be seen that the frequency of a gas-filled spring decreases with the square root of the increasing volume V_0 and the square root of the decreasing active area F of the spring. It is beneficial if the value α of the gas used is low. In other words, the constructional requirements necessary for extremely low frequencies (volume or service pressure) increase in the second power and quickly reach practical economic limits.

However, the dynamic behaviour of a spring is not determined solely by its characteristic, but also by its capacity wholly or partly to return the energy absorbed and consequently stored. Figure 3 shows a spring characteristic — as an example, progressive — in which the dynamically stressed spring is compressed from load stage (1) up to load stage (2) and thus receives an increased force ΔP ; following the explanations in figure 1, the curve from (1), taking the upper curve to (2), and the returning by (3), (4) to (1), encloses the area of energy absorbed by the spring. On release, however, the spring is displaced from (2) to (1) along the lower curve and consequently will return only the energy corresponding to the area « (2), lower curve, (1), (4), (3), (2) ». The hatched area between the upper and lower characteristics can be called the *hysteresis* of the spring. This is a measure of the energy lost through the damping effect of the spring which differs in its nature according to the different types of spring. Whilst steel torsion and volute springs have practically no hysteresis, it is very marked in rubber springs. The energy equivalent to the hysteresis is transformed into heat in the rubber spring and must be released (energy of deformation). According to the nature of the gas or vapour used, hysteresis also exists in these types of springs. In them, it can be used or reduced by constructional arrangements. Furthermore, in gas springs, it means, in accordance with well-known physical laws, that when the gas is compressed by an external force, which is accompanied by a rise in temperature, the force of the spring increases momentarily with this rise, whilst on release of the spring, it falls in proportion with the drop in temperature associated

with the expansion. A gas spring therefore has a « harder » natural action under compression, and « softer » release than a steel spring of comparable power : this is particularly marked when instead of an ideal gas, a mixture of gas and vapour (saturated vapour) is used, in which the internal humidity of the vapour produces in compression a supplementary vaporization and on release a corresponding condensation in minute drops in replacement



Fig. 3. — Difference of work of spring on compression and release hysteresis.

A. B. — Dynamische Hochstlast = maximum dynamic load. — Dynamische Kleinstlast = minimum dynamic load.

of part of the rise or fall in temperature, as the case may be. This feature of gas or vapour springs is the reason why, particularly in the case of springs with low compression, they can be used largely without additional dampers, except within the very limited ranges where they may have some resonance.

Some resonance conditions, between certain limits must be expected, even with gas springs undamped, as is shown by the oscillation conditions applicable generally to the case of a mass resting on a spring when

this mass is set in vibration by an exciting force (ΔP) with an excitation frequency (f), nearly equal to the frequency of the system itself.

Figure 4 shows, according to DEN HARTOG (6), the conductivity of the oscillations (ratio of the loading force P 1 to the exciting force ΔP) as a function of the ratios of the frequencies (exciter frequency f divided by the self-frequency f_e). It shows the advantage of operating within the «hypercritical range» for a spring-suspended vehicles; this «range» is determined by the frequency ratio and for vehicles working at varying running speeds, with variable excitation frequencies, it can reach high running speeds if the self-frequency (f_e) is kept sufficiently low. This result is satisfactory, however, only if the suspension is very soft and if possible with inherent damping, in other words the springs should have a very low value of c over the whole range of work of the springs. This explains why the designer of the vehicle, in selecting and measuring the springs of the vehicle, decides to use the softest possible system of springing. In practice, this intention, at least for traditional types of springs, is unfortunately soon limited by the fact that the softness of the springs is determined by their deflection under a variable load; moreover, to conform to certain statutory provisions (Regulations governing the construction and operation of railways and similar regulations covering tramways and automobiles), or because of restrictions imposed by the road authorities on operation, this deflection must unfortunately too often remain within moderate limits.

(b) *Requirements of light-weight vehicles as regards springs and their characteristics.*

Formerly, the necessity for giving sufficient account of the lost displacement consequent upon compression in the case of a variable pay load was less exacting than it is now with investigations into light-weight construction becoming more and more important. Moreover, in passenger vehicles, where

the quality of the suspension must be very carefully attended to, the ratios of the weights for the sprung loads already vary from 1 to 2 between the loaded and empty superstructure.

We know from technical works and patents taken out over the past ten years of the numerous efforts made to support the move towards lighter vehicles by the use of

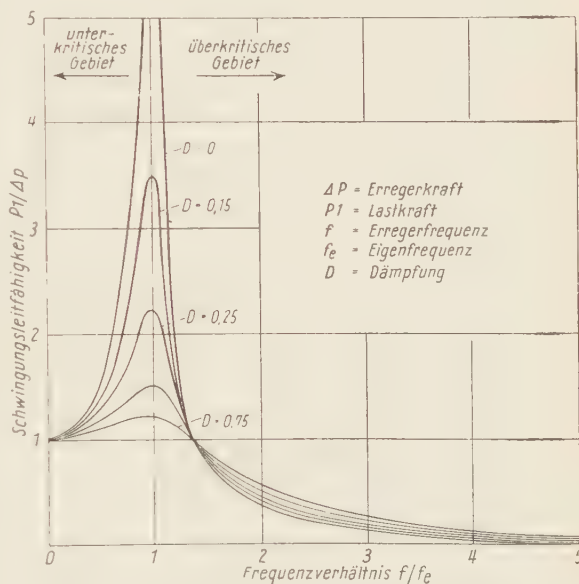


Fig. 4. — Conductivity of oscillations in function of the frequency ratios (according to DEN HARTOG).

N. B. — Frequenzverhältnis = frequency ratios. — Schwingsungsleitfähigkeit = conductivity of oscillations. — Unter-kritisches Gebiet = hypocritical region. — Über-kritisches Gebiet = hypercritical region. — Erregerkraft = exciting force. — Lastkraft = load force. — Erregerfrequenz = exciting frequency. — Eigenfrequenz = self-frequency. — Dämpfung = damping.

additional and auxiliary springs. These springs have been widely and to some extent successfully used in motor-car construction, but not so much in railway vehicles. The most simple forms of supplementary springs are the laminated springs with leaves spaced so that they become effective only when the spring has already received some deflection. Among the commoner forms, mention may be made of the double springs formed by main and auxiliary springs

(6) See before, note 5.

(for example Henschel and Büssing), the « Oltersdorf » suspension, the Kämper « roller » (Wälzfeder) type and the « double slide suspension » as well as the very recent complementary insert suspensions (type 300, Daimler-Benz cars) and similar ones ⁽⁷⁾. All attempts to solve the problem in this way by constructional means can however only have unsatisfactory results because if the springs are adopted for variable pay loads by their characteristic, there will still only be one characteristic curve presenting, it is true, bends and variable progression.

So far the most suitable devices are those by which the deflection, which varies with fluctuations in load, is automatically corrected in relation to the height and load so that the vehicle is kept at a uniform loaded height above ground level, whatever the load. These devices can be brought into operation by means of servo-controls acting between the end of the spring and the load application point so that the height of the spring lost because of increase in pay load is automatically compensated at this point. It can also be arranged so that the normal suspension, based on the empty weight, is supplemented by an additional spring with an appropriate characteristic ⁽⁸⁾ for the increased load.

Certain modern types of pneumatic springs such as those used successfully on recent special vehicles, themselves have this arrangement of different spring characteristics with progressive variation ⁽⁹⁾ placed end to end. Finally, however, it has been necessary to support — with success — the parts of passenger carrying vehicles (omnibus or small railcar) which have to be suspended, on gas-filled springs by which it has been possible to obtain a

very flat spring characteristic, well suited to each load condition between « empty » and « fully-loaded » (fig. 5); this has provided the necessary experience for constructing a spring system to cater for operation with a range of characteristics between the limiting curves of maximum and minimum load conditions; only in such circumstances is it true to speak of adapting the spring characteristic to the variable load condition of the suspended part of the vehicle.

In the graph in figure 5, a comparison is made between the characteristic of the steel (Volute) spring with a spring constant of $c = 1\,200$ kg/cm, which was formerly used and a range of characteristics for a pneumatic suspension adjustable in relation to the load, measured during tests which will be described later. This range of characteristics, the springs constants of which vary between 91 and 175 kg/cm, correspond to a spring with a « softness » about ten times greater than that of the metal spring used earlier. Because of the relationship existing for a given mass between the self-frequency of the oscillating system and the spring constant, in accordance with formula (7) :

$$f = \frac{1}{2\pi} \cdot \sqrt{\frac{c}{m}}$$

this spring constant reduced to one-tenth of its former value over the whole range of dynamic suspension has a decisive influence on the oscillation frequency and thus in accordance with figure 4, on the possibility of working at a speed much higher than the critical speed, i.e. remote from the resonance zone. In other words, a vehicle sprung in this way will be subject during operation to very small frequencies of only a few cycles (for example, small changes of ground level along the track, or rail joints and hunting on the rail). However, excitation frequencies of one Hz and below are rare and generally show only during a very short period and at low running speeds. For a suspension corresponding to the range of curves shown in figure 5, it must be expected that operation will mostly be above the critical

(7) Richard BUSSIER : Automobiltechnisches Handbuch (Automobile technical handbook). Technical editions H. Cram, Berlin, 1953.

(8) J. VAUTIER : La suspension des voitures automobiles (Automobile suspension). Swiss Technical Review, 53rd year, 26.7.1956, p. 641-643.

(9) German patent No. 845 452 : Vehicle suspension by air cushion to which the wheel load is transmitted through the medium of a compressed liquid.

range and the impulse received by the spring will be almost entirely absorbed by the spring and not transferred to the vehicle.

However, from the foregoing considerations it also results — as indicated above — that there are, in practice, limits to the

recoil effort to ensure the oscillating system holding its normal position in the absence of external impulses.

Theoretically, the ideal spring would have a constant power over the whole of its flexible range ($P = G$ and $c = \text{constant}$

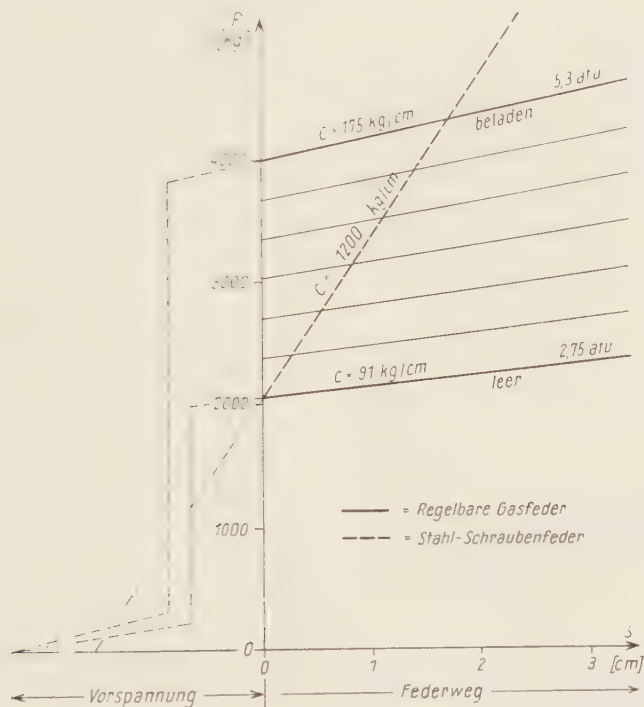


Fig. 5. — Comparative representation of a range of characteristics of an adjustable gas spring and the characteristic of a normal steel coil spring (small railcar).

N. B. — Vorspannung = adjustable tension. — Federweg = stroke of spring. — Regelbare Gasfeder = adjustable gas spring. — Stahl-Schraubenfeder = steel coil spring. — Leer = empty. — Beladen = loaded.

reduction which can be attempted in the self-frequency of a sprung vehicle. For a given mass, the frequency is only reduced proportionately to the square root of the decreasing spring constant (c); it follows that a reduction in the self-frequency of the oscillating system below a value of one Hz will require springs of extraordinary softness which, in their practical behaviour in service, could not possess the necessary

= 0) which would have to be amplified and maintained by an auxiliary designed to restore the position of the vehicle after dynamic stress. Such an arrangement is feasible but involves complications which cause heavy capital cost and even heavier maintenance; moreover, the design of the spring would, for obvious reasons, offer some difficulty.

Successful attempts have therefore been

made recently, to compile spring systems which approach the theoretically ideal behaviour without involving excessive complication. In this way, it has been possible to obtain groups of characteristics varying noticeably from those shown in figure 6.

For damping the impulses caused by the track, the spring has nearly the capacity shown by Δs ; the almost horizontal profile of the characteristic in this range demands an adjustable elasticity which, according to the static load, is adjusted to a value suitable for the particular case, between P_{min} and P_{max} . However, so that the recoil force designed to correct the position of the sprung part of the vehicle remains perfectly certain, even with a flat characteristic in the working range, there is provided above and below this range an increased value of c ; if we consider the whole working range $\Delta s'$, the spring characteristic has firstly an increasing tendency and a regressive one which after being almost horizontal (point of inflexion), in the true range of work Δs , subsequently becomes progressive.

The technical processes which allow such a form of characteristic are at present under consideration or on test in various directions. These methods, and the results of the tests will have to be the subject of supplementary articles.

III. Recent development of gas-filled springs.

(a) Evolution.

The first practical trials with gas-filled springs commenced, according to technical publications, well before 1939 with the introduction of pneumatic springing on aircraft landing gear, subsequently known as the « Faudi flexible leg ». For this purpose, a long-stroke air spring with negligible mass is particularly suitable, since it is beneficial in this case, when the aircraft lands, to have a very soft initial resilience, sharply progressive, for a reduced weight : in addition, such a spring is needed only when taking-off and landing, i.e. at long intervals, so that in spite of the use of pistons

working in cylinders, friction damping and sealing have only a secondary importance. On experience gained with these flexible legs, fittings of the same kind for land vehicles were put on the market (7). Later, and at first on motor cars, began the use of pneumatic springs in which the resilient matter was contained in flexible bellows of a type similar to those which had already proved successful in motor tyres.

One of the first proposals for a pneumatic suspension, automatically adjustable, is undoubtedly that in a German patent of 1928 (10), the pneumatic part of which was

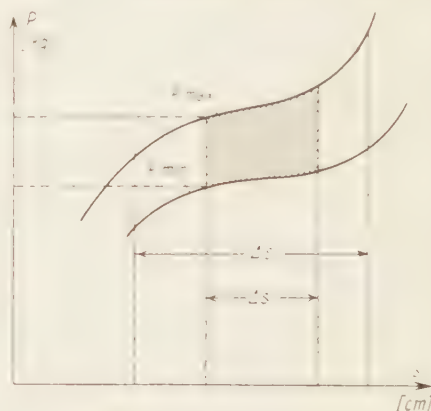


Fig. 6. — Diagram of principle showing a range of characteristics for adjustable springs (desirable form of characteristics).

a resilient spring followed by a compensating reservoir working with the spring. Between the body of the spring and the reservoir is a non-return valve with a small orifice. When the spring is compressed, the valve opens fully, on release it closes so that the air from the reservoir can flow only slowly by the small orifice. The fitting should undoubtedly have a damping effect. The whole system is kept at the maximum and minimum spring height by two pairs of electrical contacts, in the following manner :

(10) German patent No. 495 343, Class 20d, group 21 of the 13.3.1928 (published 5.4.1930) : « Pneumatic suspension of vehicles with tension automatically adjusted by the compression medium ».

With an increasing load one of the pairs of contacts closes a circuit and — by appropriately timed devices designed to prevent operation with each deflection of the spring — brings into operation an electrically driven compressor for increasing the pressure in relation to the load. When the load is reduced, the second pair of contacts closes — this time again by timed relays — a solenoid circuit which, during the period of excitation, opens a valve and allows the air in excess through the off-loading of the vehicle to flow from the compensating reservoir.

The Firestone system ⁽¹¹⁾ is well known. In this, the flexible agent is contained in a double bellows of rubber material with a metal ring in the narrow part to maintain the diameter. The body of the spring itself is connected to a compensating compressed air reservoir which, in the normal position, i.e. for straight running, operates with the air which it contains and thus makes the spring soft; on a curve the compensating reservoir is isolated by the closure of a shut-off valve working under centrifugal force and the flexible volume is thus greatly reduced so that the spring characteristic becomes progressive and clearly more abrupt, which prevents the suspension re-acting unfavourably to the action of the centrifugal force applied to the vehicle (overturning moment). For ease of understanding, figure 7 shows the principle of a spring of this kind.

Also dating from 1936 ⁽¹²⁾ is an invention relating to the suspension of motor cars by rubber pads or air cushions, in which the four suspensions of the four wheels of a car are located in cylinders connected together to balance the wheel load by a method of transferring pressure (e.g. by a liquid). This patent also includes a proposal to include in the pipe system a pump which, by regulating the pressure of gas or of air,

or the amount of liquid in the system, allows the suspension of the vehicle to be adjusted to suit the road or load conditions.

During recent years, also in the U.S.A., and primarily for motor buses, air or gas springs have been devised, supplied by the vehicle's own compressed air system and adjustable in relation to load.

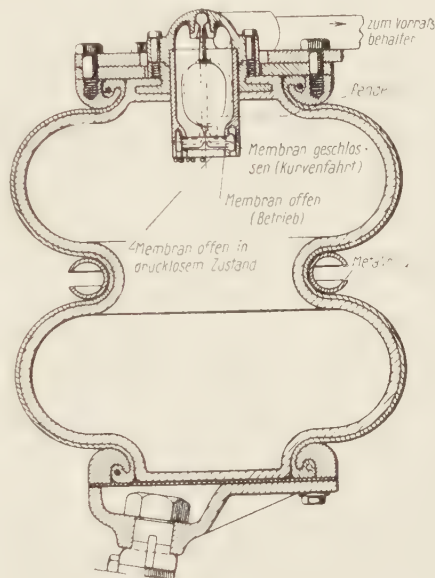


Fig. 7. — Firestone pneumatic spring (closure of the valve membrane is prevented by the vertical pendulum — in straight lines).

N. B. — Zum Vorratsbehälter = to feed reservoir. — Membran geschlossen (Kurvenfahrt) = membrane closed (running over curve). — Membran offen (Betrieb) = membrane open (operating position). — Membran offen in drucklosem Zustand = membrane open with no pressure. — Metallring = metal ring. — Pendel = pendulum.

In this case, a relatively soft pneumatic spring is used for regulating the deflection in accordance with load; where the load increases, the regulating valve controlled in relation to height supplies air to the spring from a high-pressure reservoir and, under similar conditions, allows air to escape when the load is reduced. To avoid undesirable losses of air during the normal working of the spring, the adjusting valve control is provided with retarders or chokes to prevent

(11) *Automobiltechnische Zeitschrift*, 1936, No. 8, p. 217.

(12) German patent No. 662 850, Class 63c, group 40, of the 30.9.1936 (published 22.7.1938) : « Suspension, especially for automobiles ».

too rapid an injection or release of air, at the same time allowing the load-setting of the spring in a sufficiently short space of time under the action of different conveyed loads.

These springs were developed by General Motors and used on a large number of motor buses. The system has been clearly described in technical publications, with diagrams to show the control system ⁽¹³⁾. It is also reported, in connection with the fitting and use of pneumatic springs on coaches belonging to the Greyhound Corporation (U.S.A.) ⁽¹⁴⁾ that 753 of these vehicles have run a total of 260 000 000 km fitted with these pneumatic springs and that certain of the buses with pneumatic springs supplied in 1953 have already run more than 640 000 km. Another large undertaking, the Twin City Rapid Transit Co. had in operation 395 General Motors coaches with pneumatic suspension, and over a total of 30 million km run, this company reported only one failure of pneumatic springs on front wheels per 3.3 million km and on rear wheels one per 4.9 million km. Similarly, it is reported that the Indiana Motorbus has had in service for some years motor buses with pneumatic suspension and there have been no failures, although they cover 800 km per day. Each of the 16 motor buses with pneumatic suspension of the Buffalo Transit Co. has run 130 000 km without defect. Equally favourable experience is also reported from the 100 motor buses with pneumatic suspension of the Detroit Street Railway Co., the 165 of the Metropolitan in Los Angeles, as well as 88 belonging to a Canadian firm. Engineers of the operating companies claim that reliability of these springs and the quality of the suspension, as well as economies in lubrication and adjustment make it possible to save up to 500 dollars per vehicle over its useful life.

All this information gives an idea of the wide practical use of pneumatic springs in America.

⁽¹³⁾ *Lastauto und Omnibusse*, Stuttgart, No. 6, 1.6.1955; *Omnibus Revue*, Munich, No. 11, Nov. 1952, p. 286-287.

⁽¹⁴⁾ *Omnibus Revue*, 6th year, No. 8, Aug. 1955.

In the United States, pneumatic springs have already been used on railway vehicles, as can be seen in various publications reporting their use on passenger and even on freight vehicles. For instance, it is reported that pneumatic springing has been used on freight wagons ⁽¹⁵⁾ developed by the General Tyre and Rubber Co. and the Timken Roller Bearing Co. Two-part long air cushions are used, enclosed in flexible bellows. The spring is supplied with compressed air from the brake pipe; automatic adjustment similar to that used in the General Motors suspension ensures constant height of the wagon floor above rail level in spite of the difference in empty and loaded weights. Supply to the suspension by the compressed air brake system is logical and simple. It is however, probable that in Germany, because of the safety regulations covering the fitting of compressed air brakes, it would not be permissible directly to connect the air supply for a suspension of this kind with compressed air brake pipe unless the trains had a main air reservoir pipe by which the motive unit could, without danger, constantly supply compressed air to cylinders mounted on the wagons.

The special long shape of the flexible bellows of the pneumatic springs described above is also used, according to a recent publication ⁽¹⁶⁾, on pneumatic springs of road vehicles.

Other American types generally use two-part bellows of round section, which work on the same principle as the General Motors type described above, compressed air is thus supplied, when needed — for example when the load is increased — from a reservoir fed by the compressor on the vehicle, whilst in the case of a reduction in load a corresponding amount of compressed air is discharged from the spring to atmosphere. These springs have been fitted in particular to light Diesel-powered vehicles recently built in the United States. After overcoming

⁽¹⁵⁾ *Railway Locomotives and Cars*, Sept. 1953.

⁽¹⁶⁾ Pneumatic suspension in America, Loadcraft System for Road Vehicles, *Modern Transport*, 30.6.1956.

some initial defects and after adapting the whole range of springing to the needs of construction and running of the vehicle, they have given good results.

Amongst a series of articles on light-weight Diesel trains in the United States, mention may be made of those of the General Motors Co., Electromotive Division, « Aero-train »⁽¹⁷⁾ and the « Jet Rocket » of the A.C.F.⁽¹⁸⁾.

These show that pneumatic suspension achieves a three-fold advance :

- 1) damping of shocks;
- 2) sound insulation, particularly from noises transmitted by solids;
- 3) correction of position, even when working on a curve.

The latter result is obtained, as with motor bus springs, because when running on a curve, with the vehicle body inclined outwards under the influence of centrifugal force resulting in a reduced distance between the vehicle body and the rail, compressed air is fed to the bellows on the outer radius of the curve and air is discharged to atmosphere from the inner bellows.

The Pullman Company's experimental trains, built in collaboration with Baldwin Works, and known as the « Xplorer » is also fitted with pneumatic springs of similar type, the action of which is kinematically fixed, in a perfect manner, under the body of the vehicle, linked with the guiding arrangements⁽¹⁹⁾. Lack of space makes it impossible to describe in detail this modern railway vehicle, which it may be noted, is powered by a 1 000 H.P. Maybach motor with Maybach-Mekydro transmission.

One of the most recent and most notable fittings of pneumatic springs which is worthy of mention is the bogie of the

« Pioneer III » coach of the Budd Company⁽²⁰⁾. As shown in figures 8 and 9, it now has only interior axleboxes, with disc brakes outside the wheel centre, H-shaped chassis (similar in principle to the München-Deutz bogie) ; a swing bolster, above the frame, which also serves as a compressed air reservoir, two-part cylindrical-section, on the bolster one at the right-hand side and one at the left, carrying the coach body. The quality of running obtained with this bogie under a very light coach will be watched with great interest.

Recently, too, articles have been published on principles mathematically assessed, for example an article dated March 1953 on « Pneumatic Suspension Springs »⁽²¹⁾. Based on reasoning relative to the spring frequencies to which the motor car user

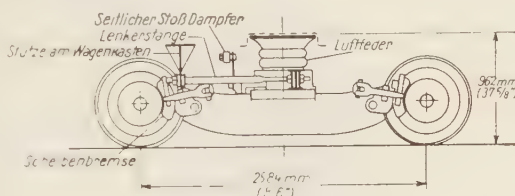


Fig. 8. — Diagram of bogie with pneumatic springs of Budd light-weight coach, Pioneer III (from *Railway Age*).

N. B. — Luftfeder = pneumatic spring. — Seitlicher Stoßdämpfer = side damper. — Lenkerstange = guide. — Stütze am Wagenkasten = body supports. — Scheibenbremse = disc brake.

is liable (figures used are 1 to 6 cycles) the author sets out calculations and opinions on spring characteristics and a method of overcoming the problem with a flexible suspension. Considering that for an adjustable pneumatic suspension a study of performance is also necessary, dealing specially with qualities of deflection and thermal control of the components, he aims to illustrate the thermodynamic considerations relative to gas-filled springs; although these considerations tend to ignore the

(17) *International Railway Congress Bulletin*, No 12 of 1955, p. 900-903, and *Diesel Railway Traction*, July 1956, p. 281-284. With photo and diagrams of pneumatic suspension.

(18) *Light-weight Train Day. Railway Locomotives and Cars*, p. 55 and seg.

(19) *First Train-X goes New York Central, Railway Locomotives and Cars*, June 1956, p. 53-62 and F. RICHTER : *Train-X takes to the Rails, Modern Railroads*, July 1956, No. 7, p. 97, etc.

(20) *Railway Age*, 23 July 1956, p. 38-42, with Special Budd Supplement.

(21) *Automobile Engineer*, March 1955, p. 112-114.

constructional conditions called for by operating requirements, they are noteworthy for their theoretical and mathematical reasoning.

(b) *Development in Europe, and in Germany in particular.*

In Europe, gas springs have been slow in finding applications. One of the first published reports (22) on the use of pneu-

sation of differences in load by a small oil pump of normal type and, when stopped, of filling the oil tank and thus let the vehicle stand with the frame grounded and the pneumatic tyres free from load. The small oil pump is driven by the motor only when the car is running; when it is not in use the suspension system is practically free from pressure so that there is little risk of defect.

In Germany, the development of gas springs was first directed towards special

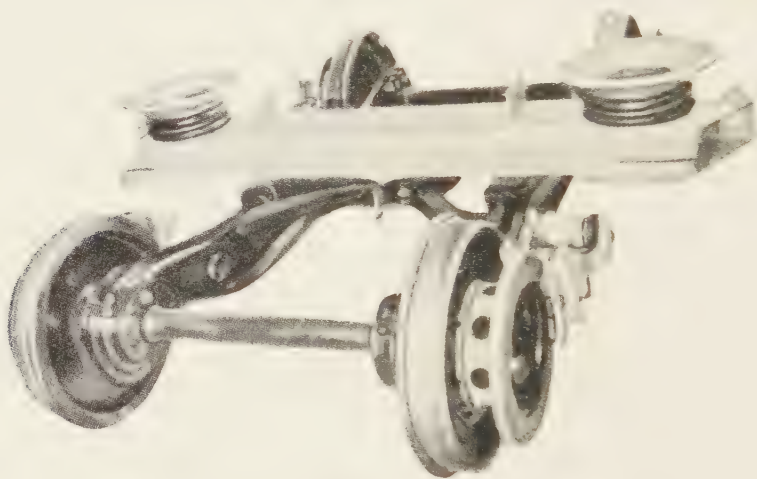


Fig. 9. — Bogie of Budd light-weight coach, Pioneer III
(from *Railway Age*).

matic springs on modern lines was in 1954 when the French Citroën Company fitted one of its cars with gas suspension and hydraulic load transmission. In this notable advance, a spherical reservoir of compressed gas cylinder was fitted under the car; pressure was transmitted to the gas by a column of liquid, in this case, oil, by which means the bodywork is carried on the frame via a compressed gas cushion. With this type of hydro-pneumatic cushion, the difficulties of avoiding leakage can be solved more easily than with gas alone, particularly when the flexible agent actuates pistons, but at the same time this also allows for compen-

requirements (wagon-carrying road vehicles) on the initiative of the German Federal Railways' Central Office in Munich, collaborating with the rolling stock manufacturers. For these vehicles, designed for road conveyance of freight wagons and heavy loads, the service required from the suspension was particularly difficult because the difference in empty and loaded weight was enormous and also it was necessary to obtain a high degree of balance of the springs between numerous wheels or numerous axles of the vehicle. For this purpose, a hydro-pneumatic suspension was developed with two load stages sufficiently soft for the vehicle when empty and when loaded, this in fact being made progressive by using

(22) *Automobil-Revue*, No. 19, 21.4.1954.

in series. As the amount of elastic agent in this spring is constant, it allowed for operation according to a single characteristic, not a series of curves. Moreover, this type of spring which had meanwhile proved satisfactory in use, and will continue to be

pression chambers of several angled axle springs, it is possible to ensure, simply and without the use of moving parts, the desired load compensation of the wheels, the vehicles thus has, even when passing over considerable irregularities, perfectly

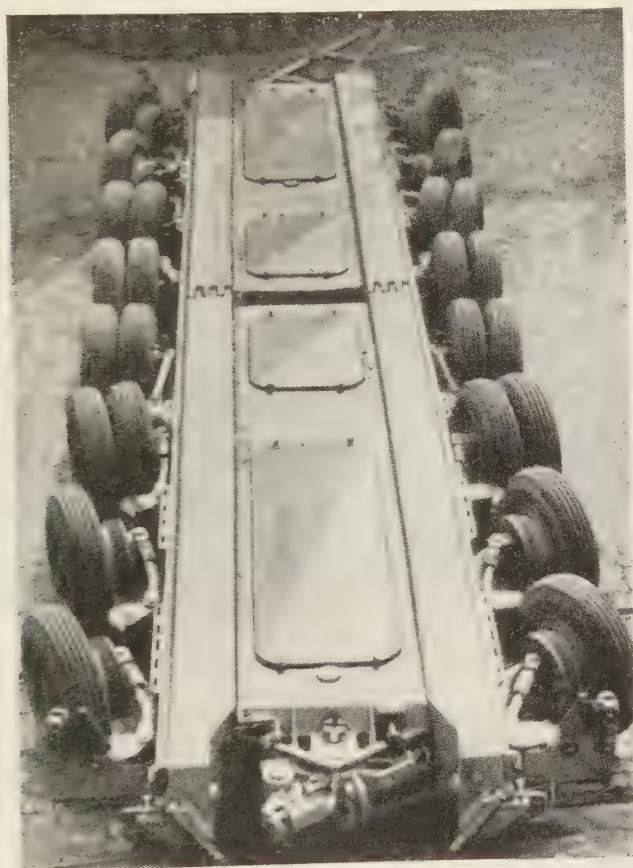


Fig. 10. — Wagon-carrying trailer with pneumatic suspension, German Federal Railways, fitted with latest type hydro-pneumatic springs (Photo SEAG).

used, allows — like the Citroën spring — compensation for the ground clearance as well as for lowering the frame. A notable feature of this pneumatically-sprung trailer (fig. 10 to 12) is that by linking the com-

stable road-holding qualities and does not experience excessive fatigue of any of the axles and springs.

Another trial was made by the German Federal Railways with a simple suspension

by pneumatic tyre during investigations into rail running gear for road-rail vehicles.

This was not an adjustable spring with a range of characteristics to allow for adaptation to momentary load conditions; on the contrary it was considered adequate to build, using a completely normal motor car tyre, a «primitive pneumatic spring» (fig. 13 and 14) which gave the prototype

tube, in which the amount of compressed air could be varied. The arrangement was very impressive. However, there has been no apparent appearance on the German market of springs of this type nor any similar models produced for practical application to vehicles.

Recently, the «Metzeler Gummiwerke»⁽²³⁾ have designed and put on test pneumatic

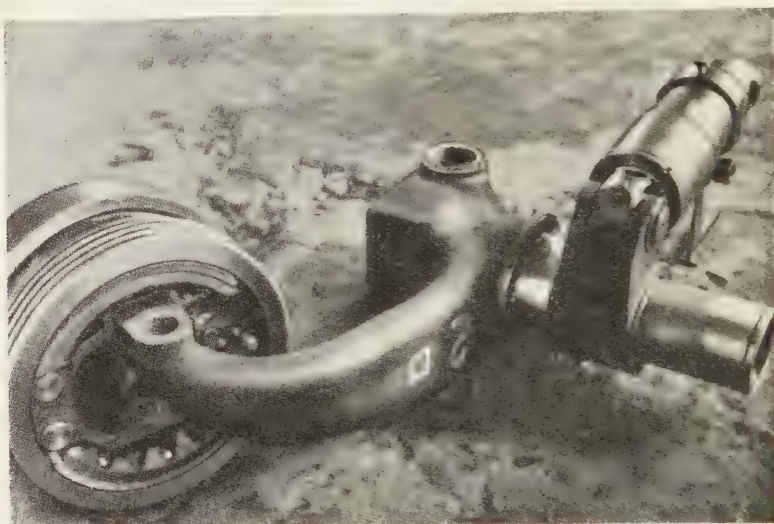


Fig. 11. — Crank axle with pneumatic spring, for wagon-carrying trailer (Photo SEAG).

with this suspension remarkably good running qualities; in addition this technical arrangement was extremely cheap as it was possible to use car tyres no longer suitable for road use.

Conditions affecting the use of a pneumatic spring on heavy road vehicles on the action which can be expected from it, have been shown to the public for the first time in Germany on the test stand of the «Continental Gummiwerke» at the International Motor Exhibition in Frankfurt in 1955, where a concrete block weighing about 5 t was made to oscillate in resonance with a simple spring formed by a round rubber

spring components of this kind, according to an announcement containing workshop illustrations, it has been possible to impose on this spring during tests on the road of more than 100 000 km fatigue stresses which a steel spring could not have supported. This case too has shown the ease with which the spring can be adapted to momentary loads and the possibility of restricting, with this spring, the inclination of express vehicles on curves. The Metzeler Company state that they can now deliver these components in two forms.

(23) *Verkehrsrundschau*, 1956, No.1.

IV. Future aims and trends in development.

Proposals by Cordes ⁽²⁴⁾ show the direction in which development is considered likely to be fruitful in Germany and is to be recommended. These deal with the use

resilient plastic or even of a space enclosed by metal to which is vulcanised layers of resilient rubber containing a gas acting as a spring. In all these proposals, there are certain spring characteristics which are progressive and to some extent beneficial, relative to the flexible part concerned and

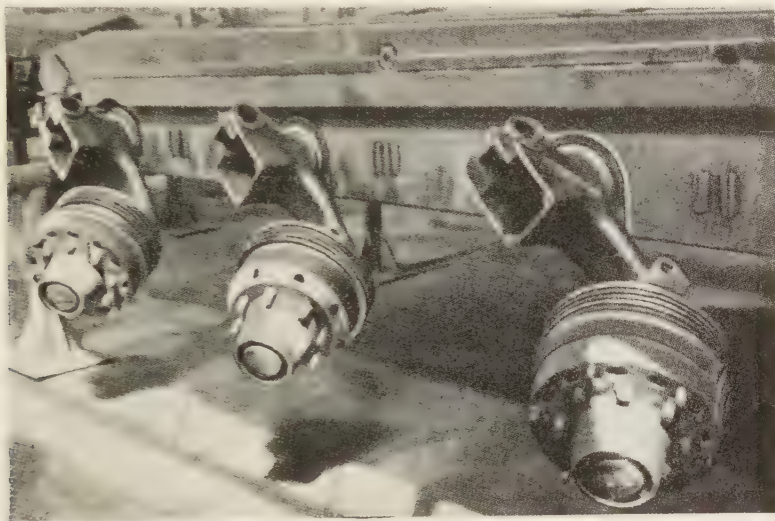


Fig. 12. — Arrangement of pneumatically-sprung crank axle in the wagon carrying trailer frame (Photo SEAG).

of air cushions, located in a cylinder in compression chambers and acting on crank arms. According to these proposals, trials with a single-axled trailer have given very satisfactory results.

Proposals have often been made to include pneumatic spring units in pivot form ⁽²⁵⁾, made up either of rubber, or of

determined by the compression of the gas which they contain. They provide, however, a single degree of elasticity, which is equally given by metal springs or by correctly calculated rubber springs.

Whilst the results so far obtained with hydro-pneumatic springs on wagon-carrying trailers have been satisfactory, the German

(24) German patent No. 811 893, Class 47a, group 17 of the 23.8.1951 : « Pneumatic spring, especially for automobiles ».

(25) German patent No. 821 886, Class 47a, group 17 of the 1.8.1950 (published 22.11.1951) : « Pneumatic rubber spring ». German patent No. 847 342, Class 47a, group 17 of the 15.11.1949 (published 28.4.1952) : « Flexible air-filled component ». German patent No. 870 620, Class 47a, group 17 of the 29.11.1951 (published 16.3.1953) : « Flexible air-filled component ».

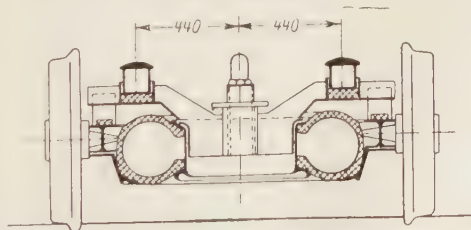


Fig. 13. — Sketch of removable bogie with pneumatic suspension, for road/rail bus.

Federal Railways have also tested, in collaboration with different rolling stock manufacturers and firms concerned with the design and manufacture of flexible bellows the possibility of creating better systems and up-to-date springs which would be free from the defects of pneumatic springs known so far. These faults, which it is hoped to eliminate, are in essence as follows :

spring. In these circumstances, the gas spring which operates on soft characteristics, determined according to a series of curves, can be used without damping. The accuracy of this argument has already been demonstrated during trials in which a vehicle fitted with very soft springs was operated without any noticeable deflection resulting from the high degree of flexibility under

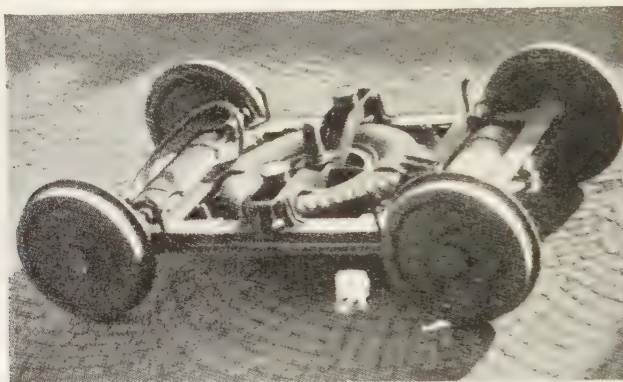


Fig. 14. — Removable bogie with pneumatic suspension for road rail bus.

1. Many of the pneumatic springs so far made have *characteristics which are very similar to those of metal or rubber springs*. Many do not in fact, make use of the possibility of obtaining a *group* of characteristics. Because of this, the true advantages of a gas spring are not provided.

2. Gas springs using *cylinders and pistons* must contend, not only with the question of keeping gas tight, but also with the *effect of friction to varying degrees*. As gas springs must be dimensioned and designed so that they have a very smooth action and allow operation well above the critical region, the spring itself has no need of damping in the ranges where, if it is not damped, there is no tendency to continued oscillation. The need for damping relative movement in the two parts separated by the spring can now only result in the type of shock impulse to be absorbed by the

the action of shock impulse caused by the track. It is obvious that in cases of this kind even slight friction would destroy the quality of resilience because it always tends to transmit residual forces, and consequently, supplementary impulses, to the suspended portion of the vehicle.

3. With the *open circuit systems* used so far, which work by charging or discharging compressed air, there is a risk of water accumulating in the body of the spring through condensation, resulting in *corrosion* and possibly leading in winter to the *formation of ice* in places where it is undesirable; unless filters are provided, which demands more maintenance, it can also produce *sedimentation* which adversely affects the operation of delicate controls. There have been scarcely any reports of this happening with pneumatic springs of the open type; at the same time, taking a

long-term view, particularly for railway vehicles, which have to have a high degree of safety and maintenance-free operation, it is essential to try to eliminate in advance all possibility of failure of this kind in the design stage.

4. *Closed circuit systems* can be built in the same manner as « open-circuit » ones if we do without the facility of adjusting the suspension in relation to the load and are not very exacting regarding the precise adaptation of the power of the spring to the load as is necessary with very soft suspensions. If a high degree of precision is required, however, the differences of pressure in the springs due to daily and seasonal variations in temperature are a difficulty in the « closed » circuit systems. If in addition, adjustability is necessary because of the softness of the springs, there are various technical possibilities; a series of them are being subjected to tests to determine their practical applicability, but it is not yet possible to give any information on the results of these tests.

5. The *costs of energy*, also require careful investigation. For short distance vehicles, particularly, the movement of passengers is considerable and in vehicles whose suspension must, as in the American machines described, have a stabilising action on the horizontal plane of the body when rounding curves, it must be expected that there will be a considerable discharge of compressed air from open circuit systems, or a notable consumption of energy even with closed circuit systems. According to the method of operation of the pneumatic spring, this consumption of energy can have a definite influence on operating costs.

6. The *problem of gas-tightness* is particularly important in closed circuit systems, and where the resilient agent is not air. Diffusion through the flexible bellows can itself lead to undesirable secondary reactions. In a general way, it is necessary to be very exacting in respect to sealing gas-springs, so that apart from the flexible bellows themselves, it is necessary to avoid the use of flexible components such as hose-pipes for compressed gas, etc.

7. The requirements which must be met in regard to *speed of recoil* are very varied. They are very severe when it is necessary to absorb efforts due to ordinary operation — as for compensation of differences of pressure when rounding curves; they also

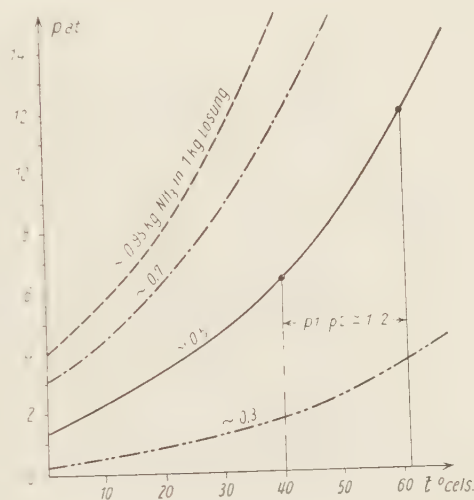


Fig. 15. — Diagram of variation in function with temperature of internal pressure of a spring filled with vapour by lines of equal absorption for NH_3 at different temperatures and pressures.

N. B. — $0.95 \text{ kg NH}_3 \dots$ = about 0.95 kg NH_3 in 1 kg of solution.

have an influence on energy consumption. When the recoil times are very short, they also offer the risk that the spring control system may become unstable; when it responds to simple deflection, the automatic operation of spring adjustment may be adversely affected when there are rapidly successive impulses, so that the spring is no longer adjustable, but has certain successions of impulses where its behaviour in service is quite uncontrolled.

8. All gas and air springs must be built for one *well-defined direction of operation*. Forces applied in another direction meet no, or very little, resistance; indeed, in the case of piston and cylinder springs, they

exert considerable guiding forces with possible jamming in the guide ways. For this reason, it is necessary to control, kinematically, with precision the direction of spring displacement by means of slides; it is however then necessary to watch carefully that the system does not set up undesirable efforts caused by friction in the joints.

Most designs and trials in progress or projected do not yet permit the publication of experience and results. These will be the subject of a later article. All that can be said at present is that along with improvements in open spring circuits, trials are also in progress with closed circuit systems using compressed air as well as other gases and vapours.

Consideration is also being given to the possibility of achieving, by simple means, the thermal regulation of these « closed » circuit systems; this appears to be important in gas springs designed for special purposes. To prove this, it is only necessary to consider some figures which are important for operating purposes. To heat the suspension of a static load of about 3 t from an assumed minimum working temperature of 30° C to that which it reaches under twice the load, or about 6 t, it is necessary if the body of the spring is filled with air (for an effective diameter of about 20 cm, a height of 25 cm and a c value of about 100 kg/cm) for example to have an electrical heating energy of about 108 000 W/min. If, instead of air, ammonia vapour (NH_3) is used, the consumption of energy for the same degree of control falls to some 8 800 W/min. If use is made of the water absorption capacity of NH_3 , which varies with pressure and temperature, to control the spring thermally, a control diagram is obtained as shown in figure 15, which represents the increasing pressure in the system with different concentrations of NH_3 solution in water. This shows that an increased temperature of 20° C alone suffices to give a spring filled with these fluids, a power which is almost doubled. Numerous other materials, which are soluble in large quantities, have a similar behaviour, as for example materials which vapourise at low temperatures, and come into the wide range of « refrigerants ».

V. First test results on the German Federal Railways.

Springs with thermal adjustment of a similar kind have been tested in service and have shown the accuracy of the reasoning given. They have also confirmed the considerations which, in the determination of the frequency to be expected from these springs, were based on the heat produced in compression of a vapour cushion which causes vapourisation of the small particles of liquid suspended in the saturated vapour, and the inverse phenomenon produced when the spring is released and the resilient fluid expands. This peculiarity means that compression and release do not follow the polytropic curve of perfect gases because with absorption and subsequent release of heat in vapourisation of the resilient fluid, the increase of pressure is slower, the value of κ being lower. In brief, these springs are softer for equal volume than pure gas springs and have a better inherent damping.

Figure 16 shows the installation of test springs of this type on a small railcar. Under the body, as can be seen to the right of one of the wheels with its axle suspension, is located the oscillating fork fixed to the frame in which is the spring with rotary bearings. In addition to the instruments necessary for the trial — the suspension also includes the regulating valve seen in the centre and designed to vary the damping effect of a choke section in the vapour circuit during pulsation when the spring is working between the cylinder of the spring in question and the compressed air receiver (boiler) forming part of the spring as shown at the top left of the figure. For the flexible spring body in this case, use has been made of flexible metallic bellows similar to the expansion links used in steam pipes and which in the figure is covered by lagging.

A thermally-adjusted vapour spring of this kind is particularly suitable when the vehicle (for example, a railcar with internal combustion engine) has large amounts of waste heat; for the rest, it operates without any mechanical drive as is necessary in air springs for compression. Alongside these advantages, there are also disadvantages :

the speed of regulation is relatively slow and cannot be increased at will; it is not possible with this spring to compensate the spring loads due to centrifugal force when rounding curves unless use is made of a fitting similar to the « Firestone » ⁽¹¹⁾ which gives a like effect; in addition, many

ticular use, these adjustable springs with their very precise technique are really suitable in practice. In this respect, it may also be considered whether some measure of combination between such springs and other types are logical and beneficial; it will be appropriate to consider, in addition

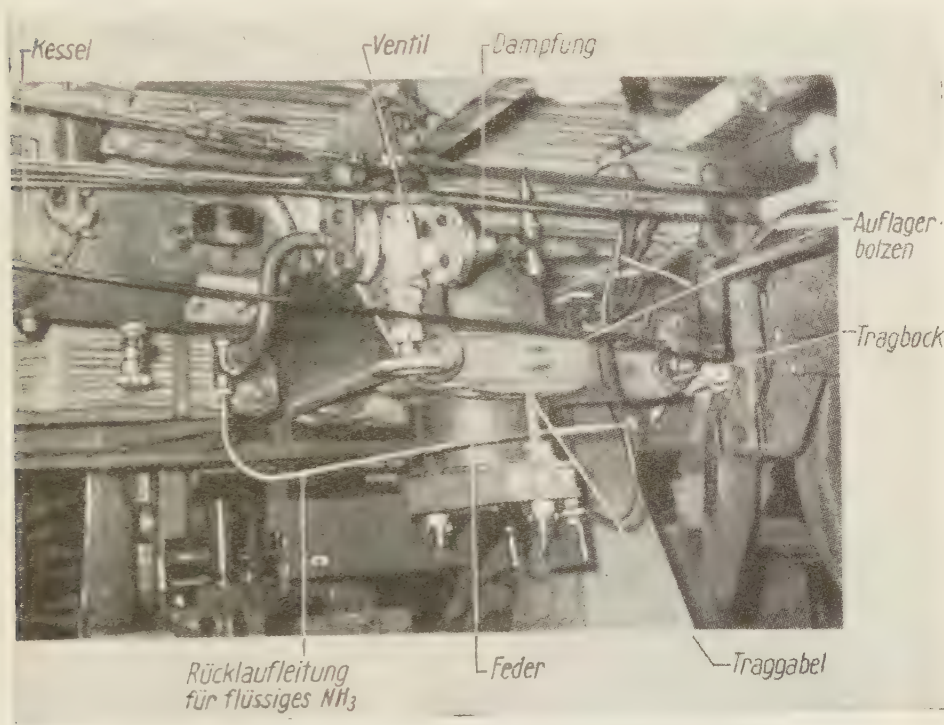


Fig. 16. — Experimental arrangement of gas and vapour springs with continuous thermal regulation, mounted under a small railcar.

N. B. — Kessel = boiler. — Ventil = valve. — Dämpfung = damping. — Auflagerbolzen = bearing centre. — Tragbock = support. — Traggabel = fork support. — Feder = spring. — Rücklaufleitung für flüssiges NH_3 = return pipe for ammonia solution.

of the materials suitable for the formation of vapour (ammonia, sulphurous anhydride, hydrate of chloral, etc) may have aggressive secondary chemical action or in the case of accidental discharge — for example as a result of leakage — may have irritant effects on passengers' respiratory organs. Only after many proposals and tests will it be possible to ascertain whether in any par-

to the neutralisation of chemical influence of the resilient fluid, the transmission of spring force by means of suitable liquids, as in the Citroën and similar devices.

The German Federal Railways, in collaboration with rolling stock manufacturers have fitted gas springs, preferably air-filled, to other small railcars and trailers as well as to a passenger-carrying trailer, for

purposes of measurement and test. Some of these vehicles are already in normal service. The experimental application of these systems of springs to main line vehicles (Express railcar VT 10⁵) as well as to the « Trans-Europ Express » is in preparation.

parts are arranged as double units on each side of the bogie frame in place of the bolster springs. Figure 18 shows the spring body supports and rollers inside the supports designed to protect the spring body from transverse stress when the bogie pivots on



Fig. 17. — Bogie, with pneumatic suspension, LFN 2 4, with spring bellows in position (Photo MAN).



Fig. 18. — Bogie with pneumatic suspension, LFN 2 4, without spring bellows (Photo MAN).

Regarding the insertion of an air spring with high flexibility between the bogie and the body of a modern four-axle passenger car (fig. 17, 18 and 19), This is a LFN 2/4 pneumatically-sprung bogie built to the design of MAN in which the pneumatic

a curve. Figure 19 shows the opposite support under the car body, the pivot and, between the two, the adjusting valve, for control of air pressure in the springs in relation to the difference in height between the bogie frame and the bottom of the body.

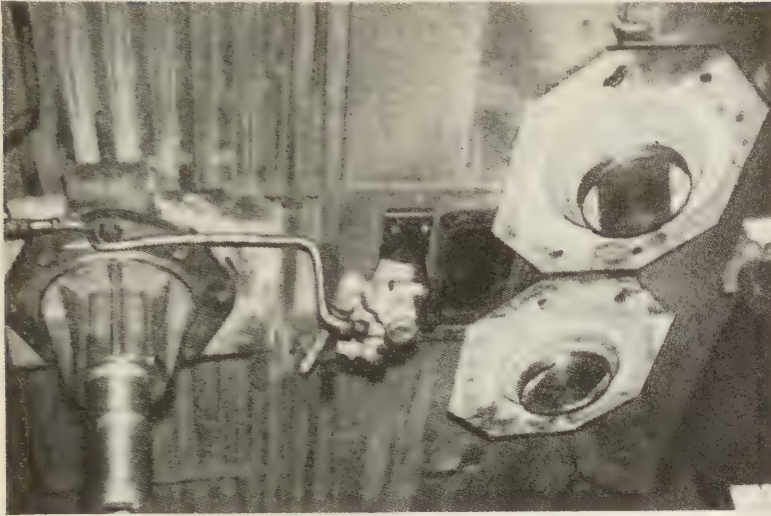


Fig. 19. — Bearings for pneumatic spring bellows of LFN 2/4 bogie on carriage body (Photo MAN).

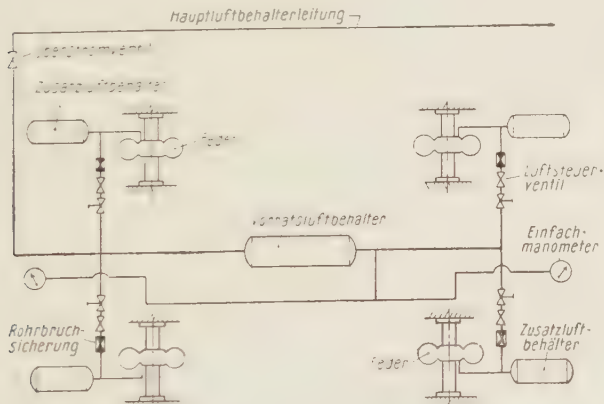


Fig. 20. — Control of gas spring with automatic adjustment to load (railbus with two motors).

N. B. — Hauptluftbehälterleitung = main reservoir pipe. — Überströmventil = exhaust valve. — Vorratsluftbehälter = air feed reservoir. — Zusatzluftbehälter = additional reservoir. — Feder = spring. — Rohrbruch-sicherung = safety device for pipe breakage. — Luftsteuer-ventil = air control valve.

The quantity of air contained in the spring body is not sufficient to give the spring the desired flat characteristic; there are, there-

fore — as shown in figures 17 and 18, by the pivot housing — compressed air receivers placed transversely between the bogie

solebars which, filled with air at the same potential, take part in the work of the springs.

Amongst the systems at present under consideration or test, there are some which provide a near approach to the ideal form of characteristic described above. With normal volume, they have already provided for a reduction of frequency from 0.7 to 0.3 cycles. These, too, will have to be the subject of a later article.

sary for the load is taken from the main feed reservoir (in the centre of the illustration), which can, as required, be resupplied by the main train pipe for compressed air. The layout of the system is relatively simple and does not include any moving parts, except for the flexible bellows, pressure control valves and air compressor which has in any case to be provided; it is adaptable within wide limits to all conditions of load. Figure 21 shows the fitting of the two

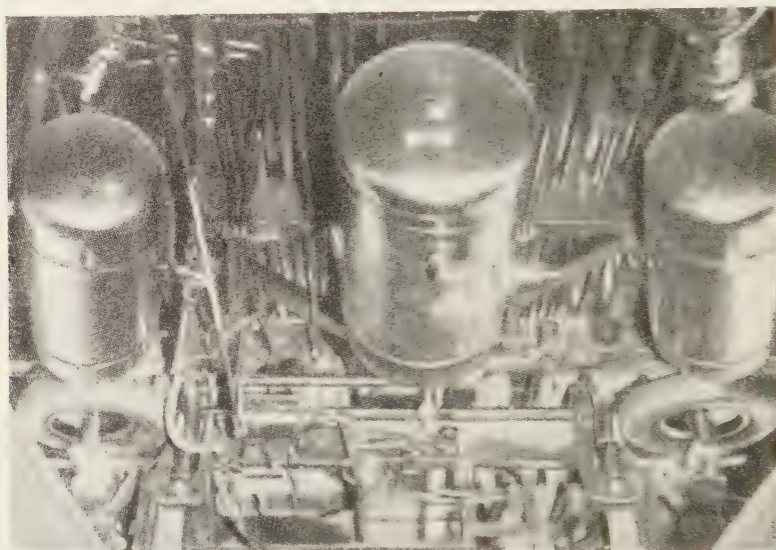


Fig. 21. — Arrangement of springs for two-motor experimental railcar
(Photo Waggonfabrik Uerdingen).

Figure 20 is a diagram of the installation and coupling-up of one of the first systems of pneumatic suspension on a small two-motored railcar, fitted experimentally by « Waggonfabrik Uerdingen ». The carriage body is supported at four points by four springs made up of single bellows of double rubber material, to each of which is connected in series an air receiver which maintains the volume of resilience. The air pressure in the space formed by the receiver and the spring is automatically adjusted by the four control valves in relation to the height of the carriage body. The air neces-

sary for the load is taken from the main feed reservoir to the right and left and the main feed reservoir in the centre, under the body of the small railcar. Figure 22 shows the arrangement of a single spring body, produced by « Phoenix » of Hamburg and figure 23 again shows the spring body during metrology tests on the bellows to ascertain their endurance.

Practical measurements have been made on vehicles fitted with gas springs, as has already been mentioned, for example on the testing plant and also during trial runs with small railcars and trailers on test sec-

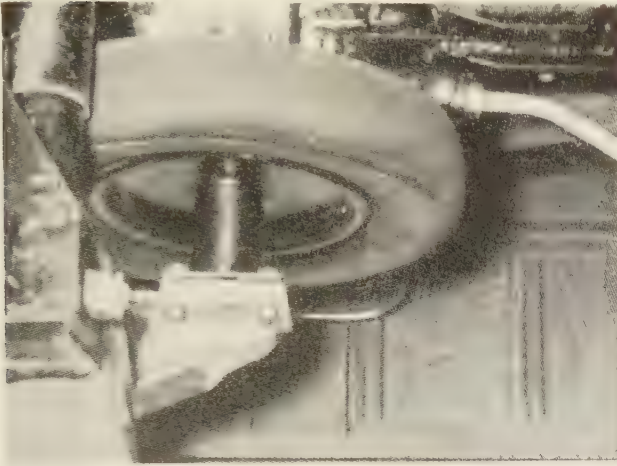


Fig. 22. — Arrangement of springs for two-motor experimental railcar (Photo Waggonfabrik Uerdingen).



Fig. 23. — Gas spring on test bench for dynamic endurance tests (Photo Waggonfabrik Uerdingen).

tions where comparisons could be made, as well as on bogie vehicles. The trials on bogie vehicles must be considered as practical tests of their suitability for this purpose, and they are still in progress.

Regarding the *trials and measurements* carried out on light railcars (Schienenomnibus) and trailers, partial results can already be given. The railcar VT 95 9569, shown in the illustrations on which the four

flexible bellows of rubber can be refilled at will with air or other gas for suspension of the body, was first tested on the test bed, during which the suspension was first studied with the car empty and with 4.5 t load with no damping and with three different dampers. The three dampers were set at 150/150 kg, 40/20 kg and 30/10 kg. In setting out the amplitudes as a function of the oscillation frequency (fig. 24), the curves correspond to a speed of oscillation of 10 cm/sec. The excitation amplitude of the rollers was + 3 mm. The vertical oscillations of the body in relation to the ground, the amplitudes of which were measured for the frequency band 0.5 to 0.8 cycles, as the laminated axlebox springs and the body springs (gas springs) were noted separately by oscillograph.

From these measurements it is immediately obvious in the region of resonance that it would be best in service, because of excitation frequencies, to avoid reliance on the suspended masses by reason of spring adjustment — it will be seen that in running without dampers maximum amplitude of oscillation occurs and that in running with the most powerful dampers there is a minimum amplitude. The increase due to the resonance of the amplitude of excitation is relatively small — between 2.1 and 2.8; on another vehicle in which the damping itself was less because of resistance to flow of air in the pipes connecting the spring body to the compensating reservoirs, this value was higher, reaching 4.6 to 6.1. The acceleration values taken at the same time in relation to the frequency (fig. 25) show a performance rather more in favour of gas springs with dampers as compared with undamped springs in the region of resonance, but are less favourable everywhere else.

The results of running stability trials during test runs with the small pneumatically-sprung railcar have been compared with those of similar tests on another vehicle of the same type (VT 95 9246) fitted with a normal steel volute spring for suspension of the body in relation to the running gear, carried out three months earlier on the same test section. This railcar was put into service with badly-worn flanges to

provide information on its performance with wheels in this condition. However, the general quality of running of the two vehicles was almost the same, and could be affected by an average rate between 2.6 and 2.9 approximately, with an oscillation frequency of 1.6 to 1.8 cycles. On transverse stability, the type of vertical suspension has only a minimum influence. It will therefore be ignored in the comparisons which follow. The differences in frequency must be

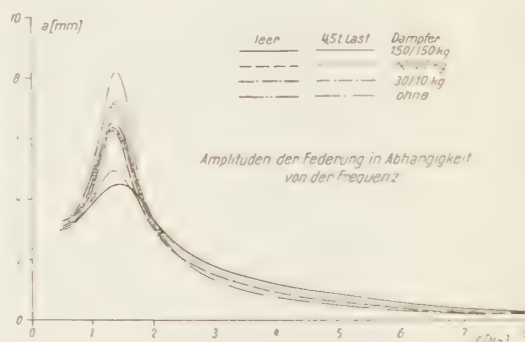


Fig. 24. — Stationary test of small railcar. Amplitude of movement of suspension in relation to frequency for different dampers and without dampers.

N. B. — Leer = empty. — 4.5 t Last = 4.5 t load. — Dämpfer = damper. — Amplituden der Federung = amplitude of movement of suspension as a function of the frequency.

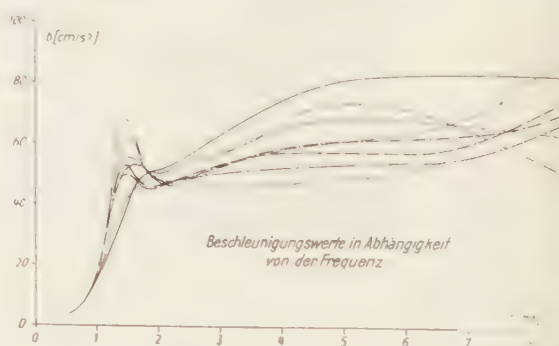


Fig. 25. — Stationary test of light railcar. Values of acceleration as a function of frequency with different dampers and without dampers.

N. B. — Beschleunigungswerte in ... = values of acceleration as a function of the frequency.

TABLE. — Results of investigations into running qualities as regards vertical oscillation of railcars VT 95 9569 with pneumatic suspension, and VT 95 9246 with metal suspension, also railcar trailer VB 98 110 with pneumatic suspension.

1	2	3	4	5	6	7	8	9	10	11	12	13
Li- ne	Vehi- cle	Type of sus- pen- sion	Load (t)	Haul- ed load (t)	Damp- er force (kg)	Speed (km/h)	Ax- le con- sider- ed	Fre- quency <i>f</i> (Hz)	Oscillation Amplitude or length <i>s</i> average max. (mm) (mm)	Standard of running quality Wz average max.		
1 2	VT 95 9569	Air- spring	4.0	0	150/150	90	Av. Ar.	2.6 2.7	1.6 1.4	2.8 2.5	2.75 2.65	3.2 3.2
3 4					60/20		Av. Ar.	2.6 2.4	2.4 1.6	3.9 3.0	3.1 2.6	3.6 3.1
5 6					0/0		Av. Ar.	2.6 2.6	2.6 1.6	4.8 2.8	3.15 2.7	3.8 3.2
7 8			150/150		Av. Ar.		2.6 2.7	1.9 1.3	4.0 2.4	3.1 2.6	3.6 3.1	
9 10					Av. Ar.		2.6 2.6	3.0 1.7	5.2 3.2	3.3 2.8	3.9 3.3	
11 12			0/0		Av. Ar.		2.6 2.6	2.8 1.4	5.2 2.5	3.2 2.65	3.9 3.15	
13 14		Gas spring		15		70	Av. Ar.	2.6 2.6	1.8 2.6	3.9 4.7	2.85 3.15	3.9 3.8
15 16	VB 98 110	Air- spring	0 4.0	0	—	90	.	2.6 2.4	1.7 1.4	. .	2.65 2.45	. .
17 18	VT 95 9246	Steel spring	0 5.0	0	—	90	.	2.8 2.6	. .	. .	3.55 3.45	4.2 4.1

Note : With regard to columns 9 to 13, lines 1, 3, 5, 7, 9, and 11 : spring movement slightly restricted by guides.
Line 13 : guides removed; line 14 : unfavourable influence of trailer with steel springs.
Col. 8 : Av. = leading axle. Ar. = trailing axle.

attributed to the variation of the oscillating mass of the vehicle in empty and loaded states.

Notes of vertical running quality, which were

taken in each case during trial running on the same section Lehrte — Isenbüttel at a running speed of 90 km/h are reproduced in the table. The running quality standards

shown correspond to the formula used on the German Federal Railways :

$$W_z = 2.7 \sqrt[10]{a^3 \cdot f^5}$$

in which the amplitude a is shown in centimeters and the frequency f in cycles (26).

The qualities noticed during the trials are assessed as under, following the experience of the Rolling Stock Testing Office of the German Federal Railways :

- 1 = very good;
- 1.5 = just below very good;
- 2 = good;
- 2.5 = just below good;
- 3 = satisfactory;
- 3.5 = fairly satisfactory;
- 4 = fit to run;
- 4.5 = not fit to run.
- 5 = dangerous.

However valuable it may be for professional appreciation by an assessment of running quality of a vehicle provided by tests and interpretation of the results, it is important to prove, with the greatest care any generalisation on the results because the assessment formula, compiled on a non-linear law, was drawn up from practical experience of *individual impressions* of the action of oscillation due to movement. In addition, the classifications do not express only the running quality of the vehicle itself, they show the running quality achieved on a set piece of track in a certain condition of maintenance; moreover, certain operating conditions such as speed, direction and intensity of wind, coupling of vehicles, etc. come into account.

Even in considering the comparative results with all the reservations indicated by the above, there is a striking difference in the vertical running quality between the two vehicles with gas springs and the one with normal type of steel volute spring. Although this reflects only an intermediate stage in the development and not a definite

condition, these results of measurements taken do tend to prove the accuracy of the arguments, set out at the beginning of this article on future types of springs and their characteristics. Mention may also be made of the differences under the influence of different dampers, the undamped vehicle having a quality of running very little below that of the vehicle with dampers (the worst values of the leading axle of VT 95 9569 are attributed to the fact that following the experimental fitting of longitudinal guides between the body and bogie frame the pneumatic spring was hampered in its oscillation). Occasionally too, it has been noticed that in a vertical direction there are harmonics of about 24 cycles but with a very small amplitude of only 0.05 to 0.02 mm probably arising from the laminated axlebox spring of the railcar.

Empty running without dampers, using springs filled with gas of SO_2 has greatly improved the running qualities as compared with air-filled springs; the improvement reached 0.2 units in the scale of classification used. The reasons for this better performance of the vapours have already been discussed. Comparison with the results of a trial run using a railcar trailer show a slight superiority in favour of the spring with a different gas used on it. The reason for this must be sought in the fact that the specific volume of air absorbing energy is 40 % greater; moreover, in the trailer the ratio of suspended load to power of the spring is less because the body is lighter and consequently the springs are subject to lower specific fatigue.

From a comparison of the railcar fitted with air springs and that fitted with steel springs, it can finally be noted that the use of a soft gas spring adjustable in relation to load, has greatly improved the quality of running. The standards of running obtained are very near to those of the best four-axled express coaches, whose vertical running quality has an average standard of 2.2 to 2.7 and in the best cases known so far reaches 1.8 to 2. The effect of pneumatic suspension on the running of the railcar and its trailer is more notable as two-axled rail vehicles do not have good vertical stability

(26) HELBERG und SPERLING : Verfahren zur Beurteilung der Laufeigenschaften von Eisenbahnwagen (Method of assessing running qualities of railway rolling stock), *Organ* 1941, No.12.

because of their design. The result of these investigations could therefore be important in the use of two-axled vehicles in future.

The first comparative trials between bogie vehicles fitted with bolster springs of the latest helical type and coaches fitted with pneumatic springs of the type shown in figures 17 to 19 were carried out at 120 km/h with sets formed of loosely-coupled coaches. Here again, as was to be expected, the values of transverse horizontal running quality were practically the same for the different types of suspension. Measurements of the vertical quality, despite a frequency of oscillation which was still much too high (1.95 cycles) gave average standards 0.1 better and maximum standards 0.4 better

with the pneumatic spring. The following results were obtained :

		<i>Steel spring</i>	<i>Air spring</i>
Average .	{ recorded }	2.2	2.1
Maximum		2.65 to 2.57	2.3

In the comparisons of these values with those for railcars, it must be noted that the speeds involved are much higher on the tests with trailing vehicles. It must undoubtedly be expected that in future it will be possible with four-axled vehicles fitted with pneumatic springs to obtain better values than those of the provisional results.

Henschel 390 HP Diesel locomotive with multiple control and rubber suspension,

by Fritz LIESE, Cassel.

(*Der Eisenbahn-Ingenieur*, No. 2, February 1957.)

Messrs. Henschel & Sohn of Cassel has built as part of its series of types several Diesel Hydraulic locomotives of the 6-wheeled design *with an engine of 390 HP*.

In designing the locomotive, the standard construction was retained in principle with the raised driver's cab already described in detail in other articles (fig. 1) ⁽¹⁾ ⁽²⁾.

1. Control for multiple unit working.

Operating programmes prescribed including light shunting have to cover the occasional transport of considerable loads which exceed the capacity of *one 390 HP locomotive* and need two locomotives to operate together. By equipping the locomotives for multiple working, two locomotives can



Fig. 1. — Two locomotives coupled for multiple unit working.

The locomotives now to be considered have a number of special features which we will describe hereafter.

⁽¹⁾ FRYDAG: « Die moderne Henschel-Diesel lokomotive » (The up-to-date Henschel Diesel locomotive). « *Glaser's Annalen* », No. 1, June 1954.

⁽²⁾ GASSMANN: « Dreischsige Henschel-Diesel lokomotive der Type DH 360 » (Henschel 6-wheeled type DH 360 Diesel locomotive). « *Glaser's Annalen* », Feb. 1955.

still be handled by one driver: in other words, the two machines can be driven from one cab.

To transmit the control impulses from one locomotive to the other, compressed air has been used. The installation was designed in collaboration with the *Westinghouse Co.*, Hanover.

It includes:

a) regulation of the engine output including the bringing into action of the hydraulic transmission;

- b) control of direction running;
- c) control of the running notches;
- d) sand-box and its control;
- e) braking.

The pipe lines required to transmit these controls are connected at the ends of the vehicles by two couplings, each coupling has four pipes. Similarly, at each end of the locomotives are the jumpers for connecting the electric lines for the indicating

singly whilst the deadman's device is cut out (fig. 2).

The control of the engine and the working of the hydraulic transmission is by means of a regulating valve « fin » fitted on the driver's table and operated by moving the driver's wheel: this valve to begin with brings into action the hydraulic transmission, then accelerates slowly the Diesel engine by a piston fitted directly to the regulator. The engine on the locomotive

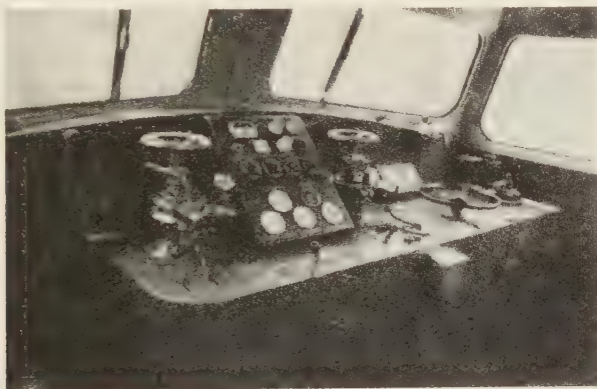


Fig. 2. — Principal driving desk.

lights of the control of running by notches and the reverser, all of which ensure the locomotives can be coupled together whatever their respective position may be. In order to have sufficient compressed air for operating the controls on the leading engine, the compressed air reservoirs are also coupled together.

In multiple working, the locomotive controlled from the leading locomotive has its brake valves, the control valves of the running gear and reverse placed in neutral: in addition, the movement of a handle closes the valves regulating the speed of the motor and those controlling the sanding which in the mid position allows air to escape; in addition, the deadman device is cut out. A device for locking the driver's control wheel coupled to this handle prevents the locomotive from being operated

with the driver can be regulated independently of the second Diesel engine and the two transmission, for example, in order to fill more quickly the compressed air reservoirs by means of the brake compressor directly driven by the engine.

To change gear and reverse valves whether from their final position or from neutral can only be operated when the double locomotive is stationary. Control lamps are provided to check the engagement of the transmission gears of the two locomotives (fig. 3).

Sanding by compressed air is operated by electric valves fitted on the leading side of the wheels in the direction of running.

The brakes in the telecontrolled locomotive operate through the automatic brake train pipe.

Apart from the control lamps of the running gear and reverse, the other devices of measure and control are not coupled together. One automatic control shuts down the engines automatically when the oil pressure falls to the lower limit.

2. Rubber suspension.

Although for a long time rubber has been used in many ways in other fields, for example, in aeronautical construction and in road vehicles, it was only rarely used as a detail in railway vehicles up to

The elastic details on the rigid framed Diesel locomotive subjected to the greatest stresses are undoubtedly the axle springs and the buffing and drawbar springs. In these applications, metal springs have always been used so far. In the locomotives now being described rubber springs for these parts have been fitted for the first time.

Rubber was selected because owing to its elastic properties and its own capacity of absorbing vibration, it is better for the purpose than steel. Moreover, the rubber spring requires no upkeep and stands up to overloads of short duration without ill effects. The silence in operation and its inherent damping property contribute to the reduction of noise.



Fig. 3. — Auxiliary driving desk: forward the levers controlling the operation of the notches of the reverser with the signal lamps.

about ten years ago. Since then the conditions have changed entirely. It is no exaggeration to say that the conditions a railway vehicle has to meet could no longer be fulfilled without using india rubber. As an example, it may be pointed out that on the standard Diesel locomotive made by Henschel & Sohn, there are some 100 parts in rubber which are used as elastic noise and vibration dampers, and notably to support the superstructure and the motor groups; some of these parts have passed their tests in service over many years already without any trouble developing even under the action of dynamic loads.

This property is especially valuable when using rubber in the suspension of the wheels and axles because the noise generated between the rail and the wheel is only transmitted to the frame to a small extent.

For the axlebox springs as for the buffing and draw springs, mixtures of natural rubber of high elasticity and great structural stability are used; the elastic properties remaining almost unchanged between the temperature of $+80^{\circ}\text{C}$. and -25°C . These springs show no appreciable hardening except at temperature below -25°C ., a temperature never experienced in Germany. This hardening becomes more marked as the temperature continues to fall to -55°C . at which it becomes « solid ».

2.1. Axlebox springs.

Up to the present the axlebox springs were plate springs carried below the body of the axlebox, and on the end of which the frame is supported through the suspension links. The axles are guided in the frame by guides bolted to the frame and between which the axlebox can move up and down. The guiding faces are fitted with bronze liners, lubricated by oil.

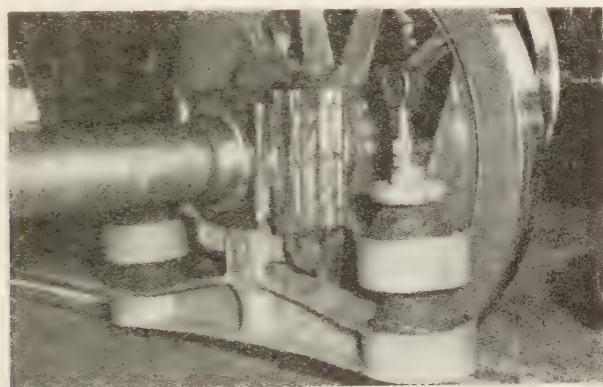


Fig. 4. — Rubber axle springing.

In this case, it was not possible to use the rubber suspension with three degrees of freedom like that Henschel had already applied to the bogies of some electric locomotives for the mining industry because the considerable longitudinal forces due to the rod drive require the guiding by the axleboxes to be as rigid as possible. It was therefore necessary to keep to the old form of guides. Nevertheless, to reduce wear the axleboxes and guides have been fitted with hard manganese steel liners requiring neither lubrication nor repairs.

Owing to the little space available on the frame of the vehicle, rubber springs with *the rubber in shear and compression* have been applied which possess, as we know, a high specific capacity of absorption of the work imposed on them. In other words, these springs can support considerable loads with relatively small di-

mensions. These elements which are hat shaped have been supplied by the Continental Co. at Hanover and have been tried out already in the springing of tramway vehicles.

By analogy with the old arrangement of plate springs, a stirrup was suspended under the box and at the ends thereof two elements in the shape of a hat were arranged in series. The weight of the frame was carried on the rubber springs by adjustable

hangers. This arrangement not only is accessible for adjustment and renewal of the springs but has the further advantage of allowing the suspensions of the two leading pairs of wheels to be equalised by an equaliser as in the case of metal springs (fig. 4).

2.2. Drawbar springs.

The volute springs used up to the present day have been replaced by nests of springs formed of a number of rubber rings separated by steel plates. These rings supplied by the Continental Co. have been used in many cases on mining locomotives of Henschel manufacture and have given satisfaction over many years (fig. 5).

2.3. Buffer springs.

Rubber buffer springs are not new and have been used for years abroad for nearly

a century on locomotives and other railway rolling stock. The rubber parts are often attached to the metal plates and differ in dimension according to the case in which they are used.

In the present instance, it was necessary to lodge in the standard D. B. buffer cases rubber springs capable of supporting the heavy loads which occur at this place whilst having a sufficiently great deflexion and a characteristic curve more or less

4. Preheating of cooling water.

To facilitate starting the Diesel engine in low ambient temperature a Webasto cooling water preheater has been provided. This can also be used either by itself or in conjunction with the normal heating by the cooling water to heat the driver's cab. All details used to control the preheater and the heating are convenient to hand on the driver's desk.

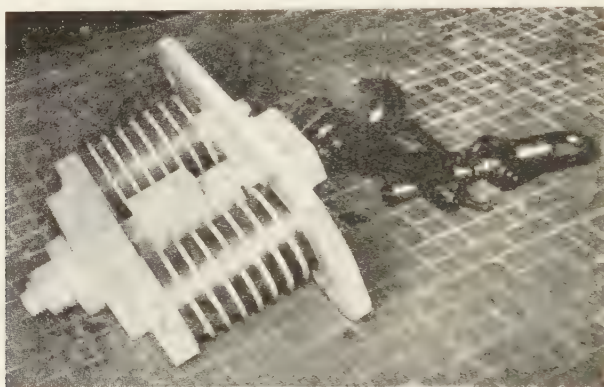


Fig. 5. — Drawbar springing with rubber rings.

identical with that of the volute springs previously used. This was done by fitting several *spring discs formed of thin metal discs covered with a thick layer of rubber vulcanised to the metal*, supplied by Messrs. Phönix of Harburg. The rubber springs too were designed so that they could be fitted with the minimum of labour to existing buffers to replace broken volute springs (fig. 6).

3. Roller bearing axleboxes.

The locomotives have been fitted with roller bearing axleboxes. In order to be able to run over curves of 90 m radius with these bearings, the middle axle has been given a play of ± 10 mm as is the usual when fitting plain bearings.

5. Measurements of the elastic deflection of steel and rubber springs on locomotive axleboxes.

Figure 7 shows that the static characteristic of the rubber spring under load corresponds to the characteristic calculated for the plate springs fitted on the standard DH 360 type.

The increased rate of deflexion of india rubber springs only shows itself when the amplitude exceeds 5 mm: for increasing amplitudes it gives marked variations.

During running tests with the locomotive with rubber springs the improvement in steadiness of riding relatively to the locomotive with steel springs was apparent even subjectively. In order to obtain precise indication on the behaviour of the two types of springs under dynamic loads, measurements whilst

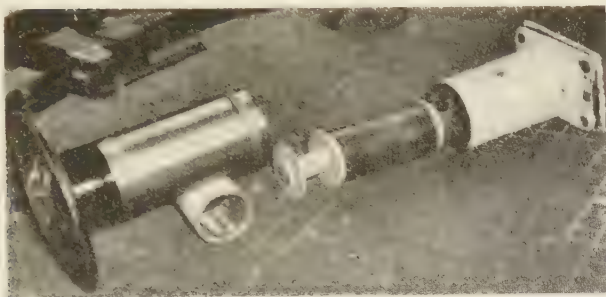


Fig. 6. --- Buffer with rubber springs taken apart.

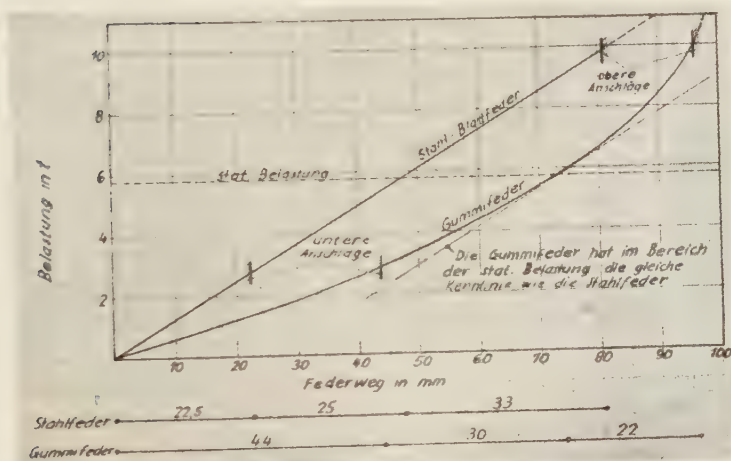


Fig. 7. — Characteristics of the axlebox springs.

N. B. — Belastung... = load in tons. — Stat. Belastung... = static load. — Obere... = upper stop. — Untere... = lower stop. — Stahlfeder... = steel spring. — Gummifeder... = rubber springs. — Federweg... = spring travel in mm. — Die gummifeder hat... = the rubber spring has the same characteristic as the steel spring in the static load field.

running were made of the spring deflexion of the boxes of the locomotives described above as well as with the steel springs as fitted to the standard DH 360 locomotive boxes.

Figure 8 shows the axlebox spring arrangement of the two locomotives. The conditions being almost identical, the results obtained could be used as basis for a comparison between the two types of springs.

Measurements of the spring deflexions were made by means of a Hottinger « W 50 » induction detector, a dynamic measurement bridge « TR 9300 » with « GM 4531 » amplifier, both made by Messrs. Phillips, and a simple direct reading recorder « FE 104 » made by Messrs. Hughes of 25 mm/sec travel and a sensitivity equal to 3. The records were taken on the part section of the line between the Cassel

Works and those at Mittelfeld. The measurements were all taken off the middle axle which is connected to the leading axle by an equaliser.

Figure 9 compares the records obtained over the same section of line at different speeds without any trailing load. All the record rolls show the points due to the rail joints, which re-appear regularly and all in the unloading part of the record. This has to be attributed to the leading

springing, all the changes of loading, that is to say, all the points at which there was a reversal of the direction of movement were noted. The result is shown on figure 10 where the changes are shown as abscissae, whilst the overload and unloading as a percentage of the static weight carried are shown as ordinates. The overloads and unloadings were all determined with the help of the characteristics of the springs according to the measured defle-

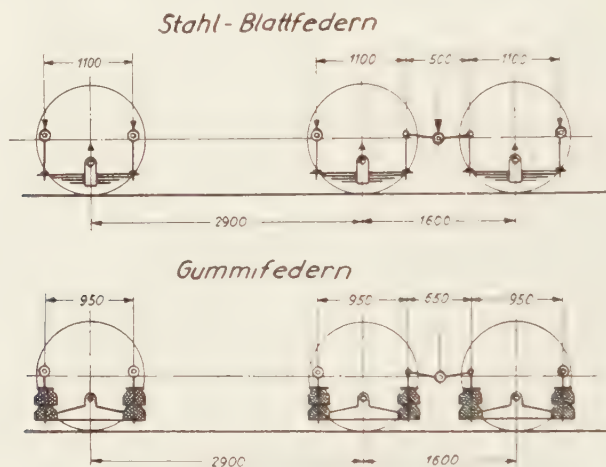


Fig. 8. — Arrangement of the axlebox springs on the locomotives used for measuring the spring deflexion.

N. B. — Stahl-Blattfedern = laminated springs. —
Gummifedern = rubber springs.

wheel transmitting all the loadings on it to the axle being recorded in the form of reduction of load through the equaliser. The result is to superimpose on the recording axle the movements of the two forward axles.

When the records from the metal and the rubber springs are compared the definite superiority of the rubber spring both at low speeds as at high is clearly revealed. This is especially so for slight shocks which are absorbed elastically by the rubber whereas the steel spring transmits them to the frame without hindrance owing to its internal friction.

In order to obtain accurate comparative values of the response of the two types of

axions; they enable a better comparison to be made than the deflexions.

Figure 10 *a* represents the curves of changes of loading at the speed $V = 20$ km (12.4 miles)/h and confirms that the rubber spring is better than the steel spring under load. In the domain of unloading the two curves coincide. This means in practice that the steel spring follows the unloading shocks imposed by the equaliser but only reacts with great inertia to the shocks due to overload of the measuring axle.

The curve corresponding to $V = 40$ km (25 miles)/h (10 *b*) confirms what we have just written. Here again the maximum of the

amplitudes of the metal spring showed itself definitely in the unloading field. Moreover the amplitudes of the two springs are much greater at this speed than at 20 km/h as a consequence of the shock impulses. A comparison between the total number of alternations shows a slight diminution in this number at the highest speed for the rubber spring and a slight increase for the steel spring. The figures of alternations however are at the two speeds higher for the rubber spring than for the steel spring.

As with the light locomotive the tests were made over a section of line 1 km long. The count of the alternations of load gave a ratio of 347 : 305 for the rubber spring which show that the differences between the two types of springs is then less felt. It must be pointed out however in favour of the rubber springs that the axleboxes of the locomotive with rubber springing had hard manganese steel guides whereas the one with steel springs had

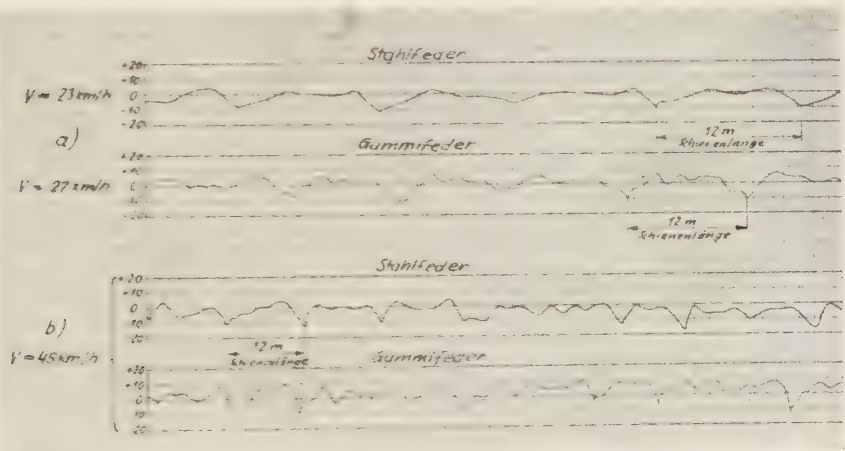


Fig. 9. — Measurements of spring deflexion on easy rising gradient without trailing load at different speeds.

+ = loading; O = position of rest; — = unloading.

N. B. — Schienenlänge = length of rails. — Stahlfeder = steel springs. — Gummifeder = rubber springs.

To eliminate fortuitous results a count was made of the number of alternations of load of all the light runs over a section of a total length of 1 km at an average speed of 20 km/h. The proportion rubber/steel rose in this case to 450 : 271 which is almost the same as the value 104 : 63 found in figure 10a.

Measurements on the locomotives running light have been completed by measurements on locomotives hauling a load in order to ascertain the effect of the friction which is due to the resulting rubbing of the axlebox guides.

bronze guides, oil lubricated. Under absolutely identical condition the rubber spring most probably would show marked advantages also when running with a trailing load or running light up a gradient.

It is also interesting to compare the oscillation diagrams shown in figures 11 a, b and c of runs over part sections with rails of different lengths. These measurements were taken with the locomotive with rubber springs alone on a line with a slight down gradient, or in other words, over which the vehicle rolled freely. Whereas with rails of 9 and 12 m

lengths the oscillation curves are more or less irregular; over rails of 8 m, there was a harmonic regularity in the form of the curve. This result is ascribed to the fact that the length of rail corresponded exactly to five times the distance between axle centres of the two leading axles.

Figure 11d represents a run over the same line in the opposite direction. This time again the regularity of the oscillation diagram is undeniable but owing to the friction in the axle bearing on the rising gradient the amplitudes are smaller.

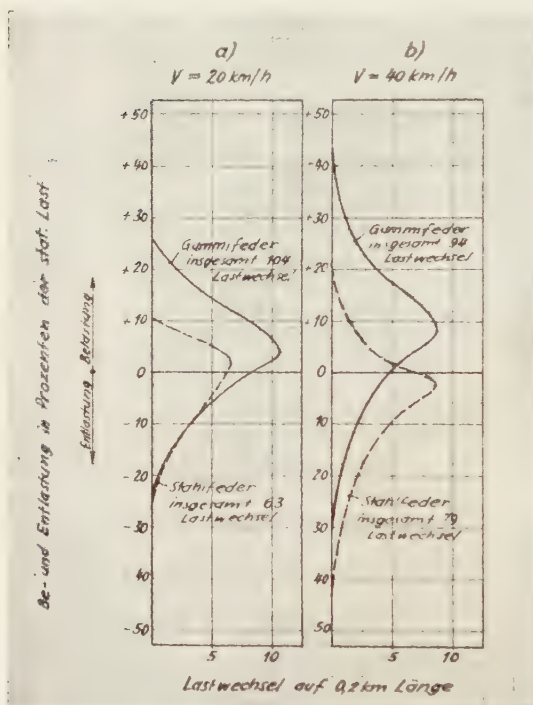


Fig. 10. — Comparison between rubber and steel springs running light at different speeds.

+ = overload; O = position of rest;
— = unloading.

N. B. — Be- und Entlastung... = overload and unloading as a percentage of the static load. — Belastung = overload. — Entlastung = unloading. — Gummifeder... = rubber spring, in total 104 alternations of loading. — Stahlfeder = steel spring. — Lastwechsel... = alternations of loading on 0.2 km length.

On curves of figures 9 to 11 will be observed for certain oscillation continuing freely that the frequency proper to the elastic system reaches the calculated value of 2.4 Hz (which corresponds for an advance of the tape of the recorder of 25 mm/sec to a length of oscillation of about 10 mm).

At the usual average speed in shunting service of 20 km/h for a locomotive with a light load the rubber spring can be said to have essential advantages over the steel springs. These advantages fall away when the load increases and as the speeds become definitely higher or lower, but on the whole they are still such as to lead us to hope that the frame and track will be appreciably better treated when rubber springs are used for axle boxes.

The locomotives with rubber springing have been in service for about a year in steel works and similar establishments working with several sets of men and have run 180 000 km (112 000 miles). According to the counts taken when recording the deflexions of the springs, this journey distance represents from 40 to 45 millions of alternations of load supported without damage by the rubber suspension. It is still not possible to predict the probable useful life of the springs but the same details have passed their tests long ago in tramcar suspensions.

Summary.

The above article describes the Diesel hydraulic locomotives of the « C » axle arrangement of 390 HP, which Messrs. Henschel have recently built. The locomotives are designed to be driven in pairs from the driving cab of one of them. The transmission of the control impulses is by compressed air.

In the standard Henschel locomotive, rubber is today largely used as a construction material. The locomotives, here described, were for the first time fitted with rubber axlebox springs and with rubber drawbar and buffer springs.

The locomotives, in addition, have roller bearings and a system of preheating the motor cooling water.

To obtain comparative values about the

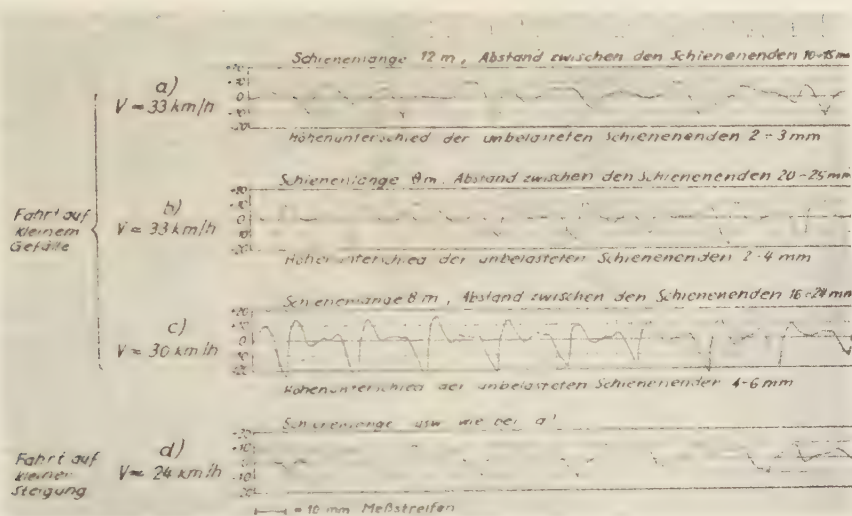


Fig. 11. — Measurements of the spring deflexions carried out with rubber springs for different rail lengths or during runs over lines on down gradients or on rising gradients.

+ = overload; O = position of rest; — = unloading.

N. B. — Fahrt. auf kleinem Gefälle = run over slight down gradient. — Fahrt. auf kleiner Steigung = run over slight rising gradient. — Schienenlänge = length of rail. — Abstand... = distance between rail ends. — Höhenunterschied... = difference of level between the ends of the rail, not under load. — Schienenlänge... = length of the rails, etc., as in a). — Meßstreifen = recorder tape.

behaviour of rubber and steel laminated springs under dynamic loading, measurements were made of the deflexions of the springs on the locomotive axleboxes of the locomotives described and on those of the comparable standard type DH 360 with

steel springs. These showed that the rubber springs improved the riding of the locomotive clearly and especially by *absorbing the slight shocks elastically*, which suggests the frame and track will be less stressed.

Oil fired steam locomotives,

by Walter DONNER. Ing. Cassel.

(*Der Eisenbahn-Ingenieur*, No. 4, April 1957.)

The financial difficulties of the German Federal Railways after the second world war obliged them to take very extensive rationalization measures in all administrative and operating fields. In addition to the urgent reconstruction of the permanent way, the safety installations and bridges, the creation of a powerful stock of motor units came up against particularly serious difficulties. First of all, they had to limit themselves to restoring to order the existing steam, electric and heat engined locomotives, which were often badly damaged, as well as the railcars.

It soon became apparent however that on the one hand the numbers of such units that were repairable were insufficient for the rapidly increasing traction requirements, and on the other that the steam locomotive in particular, because of the long technical operations involved and the amount of its costs in servicing and fuel, did not always line up with the requirements for an up-to-date motor unit. The results of studying this problem, as well as the favourable operating results obtained by other Administrations with motor units, showed indisputably that rationalisation of the traction department would mean a change in the structure of the locomotive stock of the Deutsche Bundesbahn.

After a certain train loading on the line, electric traction alone is from every point of view the most economic and the best. This is why the Deutsche Bundesbahn has prepared a vast electrification programme and since the second world war has already considerably extended its electrically operated system. In the same way, for passenger services on secondary lines and local shunting services, the use of the steam locomotive is indefensible today for

economic reasons. For these services, the German Federal Railways have developed the use of small railcars or Diesel locomotives driven by one man, thereby undoubtedly considerably reducing their traction costs.

On the other hand, it is not yet sufficiently proved that the numerous heavy steam locomotives which are not in any way worn out should be replaced by equivalent Diesel locomotives on the non-electrified lines for hauling passenger and freight trains. In view of the financial position of the D. B., which is not at all satisfactory even today, it is essential to find the most suitable and most economic motor unit for these jobs.

In the following article we are going to examine if, and to what extent, fuel economies could be counted upon by replacing steam locomotives by Diesel locomotives for the passenger and heavy freight services on main lines.

First of all, we will compare the operating results obtained in November 1956 with a series 01¹⁰ (coal fired) steam locomotive and a V 200 Diesel locomotive (the values used are taken from the Bema 2 statistics).

a) Mileage per day of use of the locomotive :

01¹⁰ = 691 km;

V 200 = 750 km.

b) Loads hauled :

01¹⁰ = 530 t;

V 200 = 339 t.

c) Fuel used per million gross t/km :

01¹⁰ = 28.20 t;

V 200 = 6.940 l.

It must be remembered that in the load figures given under b) above, the weight of the engine is included ($01^{10} = 180$ t; $V200 = 80$ t). To obtain a pertinent basis of comparison, the above fuel consumption figures given in c) must be related to the 1 million coordinated gross t, km; they then amount to:

in the case of the 01^{10} :

$$\frac{28.20 \times 530}{(530 - 180)} = 42.70 \text{ t}$$

in the case of the V200 locomotive:

$$\frac{6940 \times 339}{339 - 80} = 9084 \text{ l.}$$

Starting with the above figures, with coal at 63 DM per ton at the mine, and Diesel oil at 36 Pf the litre at the wagon, we get the following specific consumption figures:

locomotive 01^{10} :

$$42.70 \times 63 = 2690.10 \text{ DM per million coordinated gross t/km;}$$

locomotive V200:

$$9084 \times 0.36 = 3270.24 \text{ DM per million coordinated gross t/km.}$$

This summary calculation shows that the fuel costs for the V200 locomotive are about 21 % higher than those of a coal fired 01^{10} locomotive.

Both these two types of engine are driven by two men; the use of the V200 therefore does not lead to any saving in driving wages. The cost of buying a Diesel locomotive is certainly much higher than that of a steam locomotive of equal power. As regards the useful life and maintenance costs for heavy Diesel locomotives, the D.B. has not yet sufficient experience to give any figures. But it is difficult to see why the Diesel locomotive should be better than the steam locomotive in this respect. We must not forget to include the considerable sums expended when putting vehicles with thermal engines into service on making

new and costly installations for preparing these machines for work.

The Diesel locomotive undoubtedly has certain advantages over the coal fired steam locomotive, some of which we will mention: cleanliness of working without smoke or soot, longer periods of user owing to the reduction in the technical preparation times and the suppression of turning round at the terminus stations of the run, no danger to forests, embankments, buildings, etc., from sparks and cinders, etc. But taking all in all, it does not seem possible to prove indisputably that under present conditions the replacing of steam locomotives by Diesel locomotives has definite economic advantages.

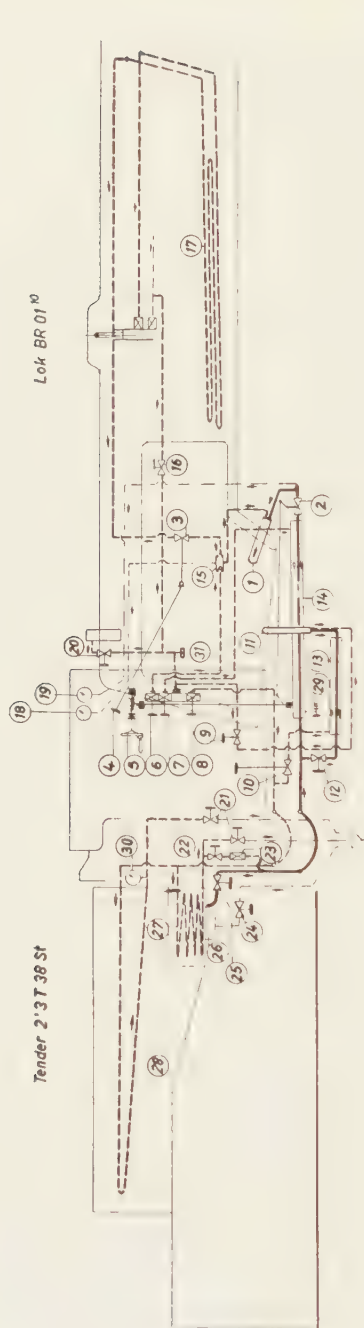
The comparison is still more in favour of the steam locomotive if the latter instead of being coal fired is equipped for oil firing. The D.B. has so equipped an 01^{10} express locomotive; we will describe the boiler and give details of the interesting operating results obtained.

I. Description of the main heating by oil for steam locomotives.

When converting a steam locomotive that has previously been coal fired, the boiler and tender do not have to have any great modifications made to them. The parts designed for burning coal are removed, so as to replace them by the corresponding components of oil firing (see fig. 1).

This does away with the grate, the ash-pan, the device for wetting the ashes in the ash-pan, the device for water spraying the smoke box and the spark arrestor. The firebox arch remains unchanged. It is only necessary to modify the chimney and nozzle. The chimney is fitted with a sleeve which reduces the diameter of the opening from 580 mm to about 510 mm and the minimum diameter from 490 mm to about 465 mm. The nozzle is reduced from 155 to about 145 mm diameter.

No modification had to be made to the tender for the installation of the oil tank and additional equipment required for oil firing, apart from reducing the web of the upper angle iron used to stiffen the coal bunker and the removal of the inclined back of this bunker. After the oil tank is removed, the coal bunker can therefore be used again for coal firing.



- | | |
|--|--|
| 1. Main burner. | 21. NW 10 regulating valve. |
| 2. Valve for regulating the oil. | 22. NW 16 steam valve for blowing out the oil main pipe. |
| 3. NW 16 superheated steam valve. | 23. NW 16 check valve. |
| 4. Control wheel for the superheated steam valve. | 24. Blow down valve of sediment collector. |
| 5. Wheel of oil regulating cock. | 25. NW 40 oil regulating valve. |
| 6. NW 16 steam valve for the main burner. | 26. Sludge collector. |
| 7. NW 16 steam valve for the oil preheater of the main burner. | 27. Coil. |
| 8. NW 16 steam valve for the tender heating coil. | 28. Oil tank. |
| 9. Needlevalve for regulating the steam of the secondary burner. | 29. Automatic device for draining condensation water by stepped nozzles. |
| | 30. Telethermometer of oil tank. |
| | 31. Connections for taking steam from an outside source. |

Diagram of the oil firing of locomotive BR 0110.

The completely *welded oil tank*, of 13 m³ capacity, is mounted in the coal bunker of a 2'3 T 38 tender from which it is detachable. Oil is taken in at the back of the tank. It has a perforated sheet strainer. The cover is kept tight by two wing nuts. To avoid heat losses, the cover of the tank is lagged. There is an air vent in order to prevent the setting up of a pressure in the tank.

The oil used (Bunker C oil) being very viscous at a temperature of 15-20°, two coils of heating tubes without welding have been provided in the tank: the upper coil is intended to heat the whole of the contents of the tank, while the other, smaller, coil is fitted immediately above the elbow through which the oil runs. The two coils, which have a total heating surface of 2.285 m² are supplied with steam through a connection formed of a rubber pipe and metallic tube ducts. The removal of the condensation water formed by the heating steam can be regulated by means of two valves. To enable the temperature of the oil to be controlled, there is a telethermometer on the front of the tank.

The fuel oil passes onto the locomotive through a stop cock on the tank, piping and a flexible connector. To collect sludge and condensation water, there is a sump under the front part of the tank with a drain cock. The steam required to heat the oil is taken from the rear steam fountain to a special distribution steam fountain through a stop cock. In the conduit, between the stop cock and the distribution steam fountain there is a threaded pipe with a closing device, enabling connection to an outside steam supply for heating. When the locomotive is lighted up, steam produced from some outside source can be introduced through this connection in order to run the oil heating equipment and the auxiliary blower. The blower conduit is connected to the outside steam connection by a pipe and stop cock. To allow the rapid opening of the auxiliary blower, the valve of the standard blower has been replaced by a rapid opening valve; to operate this valve, a lever has been provided in place of the wheel.

A special steam fountain is fitted with three stop cocks which control successively from top to bottom the pipelines supplying steam to the burner, the oil preheater, and the heating coils of the tender. On the left side of the locomotive, between the tender and the burner, near the firebox foundation ring, another preheater has been fitted with a heating surface of 0.96 m². The oil is brought up there to a temperature of about 120° C., which is necessary for vaporisation. To regulate the escape of the condensate, an automatic drain cock has been fitted behind the preheater.

The amount of oil required at any moment to feed the locomotive is regulated by means of a turnable valve arranged behind the preheater. From there, the oil flows directly to the main burner under the firebox supports. The burner is made of cast iron. The oil flows through the opening above the steam jet; it is picked up and vaporised by the steam. The main burner is only linked to the oil and steam conduit by a strap; this latter has only to be loosened for the burner to be easily and quickly removed for cleaning.

To vaporise the oil in service, superheated steam is used. The steam required for this purpose is taken from the steam conduit of the front steam fountain and sent through a stop cock into a separate superheater element of the steam collector. The superheated steam passes through a regulating valve, the control wheel of which is fitted beside the wheel of the supply valve in the driving compartment beside the fireman. Beyond the regulating valve, there are three branches in the feed to which the steam conduit for feeding the burner by steam from an outside source is connected, or by saturated steam when steam is being raised in the locomotive, as well as a pipe to the pressure gauge. The pressure gauge fitted near the telethermometer on the front of the firebox enables the driver to check and maintain correct vapourising pressure. In this three branch connection, there is a ball check valve which shuts off the branch which is not

required according to whether the burner is to be supplied with superheated or saturated steam.

In addition to the main burner, there is also a so called permanent or lighting up burner which as its name indicates has two important jobs to carry out: it must in fact:

- 1) when running with the regulator closed, when stopped in stations and when stationary or being prepared at the depot, produce the steam required for the auxiliary installations of the locomotive (compressor, lighting dynamo, etc.), and

- 2) make it possible to light up the main burner when this has been out of use for some time, even if the refractory bricks of the firebox have got cold.

To protect the firebox from the direct action of flames, it is lined in the lower part with refractory bricks. This masonry gets red hot in service and thus facilitates the lighting up and burning of the vaporised oil. The ash-pit is replaced by a firebox also lined with refractory bricks. The combustion air penetrates partly through an opening in the front wall of the firebox where the burner is fitted, but mainly through openings in the bottom of the firebox. In the rear part of the bottom of the firebox, a large cleaning trap has been arranged, which can be moved and locked from the driving compartment.

Under the bottom of the firebox is the air space with shutters in front and at the sides, which are closed and opened during running by means of control rods beside the fireman.

The firebox door is replaced by a special type of door with an inspection window and variable air admittance vanes. For protection against excessive radiation of heat, it has double walls and special insulation in addition. An air duct is connected to this door which extends under the floor of the driving cab for a secondary air supply; it is also a protection against possible flames flowing back when lighting up if the firebox door is opened.

II. Working of the oil firing in service.

a) *Raising pressure in the boiler.*

An outside source of steam is necessary to get the cold locomotive under pressure. For this purpose any locomotive in steam can be used, the steam being taken through the steam heating pipe. Before the burner is lighted, the temperature of the oil in the tender must be raised to about 50° C. If the tank is full, it takes about 3 h to heat the oil from 20 to 50° C. As soon as the temperature is high enough, the burner is lit by means of a torch (cleaning rag impregnated with oil) introduced into the firebox through the burner opening. After lighting up, the flow of oil is reduced and only the amount strictly needed for uniform, smoke free combustion is allowed to run in. The heating up of the boiler takes about 30 min to the start of steaming. When the pressure in the boiler reaches 5 to 6 atm. the outside source of steam can be cut off and the locomotive works on its own with its own steam.

b) *Before starting off.*

When stopping, the increase of pressure in the boiler takes about 2 min per atm. This figure obtained by experience gives the engine men a means of regulating the main burner in time for the pressure to be raised to its normal value prior to the departure of the train. If the pressure reaches 16 atm. before departure, the main burner must be cut out and the lighting up burner run.

c) *During running.*

After departure, as soon as the locomotive exhaust is assuring sufficient air, the auxiliary blower must be closed. The oil tap is regulated in such a way that the production of steam corresponds to actual consumption. The fireman must carefully note the degree of admission and pressure at which the driver is working. The driver can usefully draw the attention of the fireman to any important variation in the work supplied by the engine, as well as whenever

he closes or opens the regulator. In every case, care must be taken to avoid making too much smoke, which always means a waste of oil. Good combustion, with the proper amount of air, can be recognised by the slightly brown colour of the exhaust gases; if the gases are completely colourless, it means that there is too much air.

The oil fired locomotive also has a steam tube cleaning device, which makes it possible to clean the tubular plate of the firebox as well as the large and small tubes during a difficult run on a suitable section of the line where no one will be inconvenienced by the smoke.

d) *After running.*

The oil channels must be blown through during a service stop. Though this operation can be done at any time in the case of the tank stop cock, in the case of the burner, the oil cock and the preheater it should only be done after the masonry of the firebox has cooled down, to avoid the flaring up of the oil residues thrown out of the burner during this operation.

When the installation is put out of work, the first step is to close the turning valve and the tank stop cock, and then the corresponding steam valves. After shutting off the burner, the blowers and all the air

intake vanes of the heating installation must be closed as in order to prevent as far as possible the penetration of cold air into the firebox.

III. Operating results and results of trials to date.

During the summer of 1956, the 01 1100 locomotive at Cassel shed was equipped with an oil firing installation designed by the firm of Henschel & Sohn. On the 10th July 1956, the locomotive was lit up for the first time. The amount of smoke was considerable at first. A few immediate improvements to the regulation of the oil flow and air soon reduced this defect to a reasonable proportion, so that the locomotive testing department of the Bundesbahn Central Office at Minden (Westphalia) was soon able to check the working. The observations made during this period showed the need for further modifications, which the firm of Henschel & Sohn carried out by the 31st July 1956. From the 1st August, the locomotive was put into service in link 1 (express) from Cassel shed.

A comparison of the operating results obtained with the oil fired locomotive and the 7 coal fired locomotives of the same link in August and September are given in the following table:

Month 1956	Locomotive No.	« Run » 10 km	10 ⁵ gross tkm	Consumption of fuel in t per		Remarks
				1 000 km	10 ⁶ gross tkm	
August	01 1100	22 043	11 563	10.66	20.33	* The trial locomotive ran during the first two weeks of the month with too small a sump at the left valve; in addition several types of burner were tested during the month. These explain the increase in the consumption.
	Average for 7 other locomotives	20 925	11 233	14.38	26.82	
Sep- tember	01 1100	22 084	10 681	11.28*	21.22	
	Average for 7 other locomotives	19 426	10 370	14.62	27.42	

With a price for coal of 63 DM per ton at the mine and for oil of 95.5 DM per ton at Hamburg Port, we get from the consumption figures for August 1956 the following specific fuel costs:

Coal firing:

$$14.38 \times 63 = 905.94 \text{ DM/1 000 km run.}$$

Oil firing:

$$10.66 \times 95.5 = 1 018.03 \text{ DM/1 000 km run.}$$

The fuel costs per 1 000 km (620 miles) run by the oil fired locomotive were therefore higher than those of the comparable test locomotives by $1 018.03 - 905.94 = 112.09$ DM, i.e. 12.3 %.

But if we take the cost of American coal.

which is already being imported in fairly large amounts, i.e. 88 DM per ton at port, we get for the test locomotives:

$$14.38 \times 88 = 1 265.44 \text{ DM/1 000 km run, or 24.3 \% more than for the oil fired locomotive.}$$

When it is remembered that the oil firing equipment is something entirely new for all the establishments of the Bundesbahn concerned, it can be stated that the operating results it makes possible are excellent. After being in service for some time, the oil fired locomotive was handed over to the Minden locomotive test department for trials to be carried out. As the following extract from the results given by certain trials shows the performances realised were excellent:

	80	80	60	100
Speed km/h	80	80	60	100
Pressure in the steam chests . .	15.2	14.8	15.2	15.2
Admission %	30	35	40	27.5
Power at drawbar HP	1 920	2 100	2 050	1 870
Total steam consumption . . . kg/h	13 450	14 250	16 000	14 400
Temperature of the steam . . . °C	447	454	451	451
CO ₂ contents of combustion gases %	12.9	12.9	11.2	13.4

These performances obtained from the oil fired locomotive without undue strain on the crew are appreciably better than those required in service of the BR 01¹⁰. They can only be required of a coal fired locomotive for a very short period, because the physical strength of the fireman is excessively tried.

IV. Maintenance.

After a period in service of about four months, no defects were found in the fire-box of the 01 1100 locomotive which could lead to fears that the material of the fire-box could not stand up to the much greater thermal fluctuations than are found with coal firing.

In general, the superheating of the steam is carried to 60 to 70° more on the oil fired locomotive than on the coal fired locomotives. Temperatures of 430 to 470° C. are reached. These have not caused the slightest difficulty so far. If needs be, the superheat can be reduced by suitable reduction of the superheater elements.

On the oil fired locomotive, the frequent maintenance work on the ash pit and smoke box, as well as the considerable cost of replacing the bars of the grate is done away with. It is not yet possible to foretell to what extent these savings will be wiped out by maintenance work on other parts of the oil heating equipment. It must be admitted that the maintenance costs of the two types of heating will be about the same.

V. Data concerning the fuel (« S » crude oil).

By « crude oil » is to be understood all the liquid fuel resulting from coking of solid fuels at high or low temperatures, or obtained as a residue from the treatment of crude petroleum. The greater part consist of residues produced during the manufacture of petrol, paraffin, Diesel oil, lubricating oil and bitumin from crude petroleum, which is given the general name

of mineral heating oil. According to the amount of crude petroleum in different hydrocarburant groups, oils are obtained which differ one from the other from the point of view of calorific power, specific weight, point of solidification and viscosity. The crude petroleum from the Middle East, which are those mainly treated at the present time, mostly have a large paraffin contents and consequently give oils with a high point of solidification. The character of the crude petroleum varies according to the place of extraction, so that the analysis data of the oil extracted always show certain fluctuations.

According to the viscosity and point of solidification, a distinction is made between very fluid oils (L oils), oils of average fluidity (M oils) and oils of small fluidity (S oils). The latter correspond to the No. 6 Fuel oil of the ASTM-Standards and are also called according to international classification « Bunker C oil » or « S oil ». The S oil costs less than the L and M oils, but owing to its high viscosity it needs large installations for heating and combustion; its use is therefore essentially justified in the case of large industrial installations as well as ship and locomotive boilers.

The German standards (DIN 51 603) lay down for S oil:

viscosity 40° Engler at 50° C;
minimum calorific power 9 200 kcal/kg;
(in general 9 600 to 9 700 kcal/kg)
ignition point. at least 65° C.

As regards the point of solidification and sulphur contents, the DIN 01 603 standards give no values for S oil. As regards its chemical composition, the following average values can be admitted:

carbon up to 0.1 %;
hydrogen 1.5 to 3.5 %;
oxygen up to 1 %;
sulphur 10.5 to 11.5 %;
ash 85 to 87 %.

Its specific weight amounts to 0.96 to 0.97 at 15° C. To get good vaporization, with S oil it is necessary for the viscosity to be 2 to 4° Engler. The oil must therefore be heated to about 120° C before it reaches the burner.

VI. Summary.

The results obtained to date with the 01 1100 locomotive in regular service and during trial runs with the dynamometer make it possible to state already that the oil fired locomotive is not only better from the point of view of output and economy than the coal fired steam locomotive, but also than the Diesel locomotive. The oil fired locomotive can be lighted up and put under pressure in a space of time unbelievable so far with steam locomotives, and raised to the maximum pressure of 16 atm. in 1.5 h or so. The time taken to carry out the work on return to the sheds is also lower, because there is no grate or smoke box to be cleaned out. The fuel (oil) only has to be taken on at longer intervals, generally after some 800 km (500 miles), i.e. after what is known as a complete utilisation of the locomotive. The result is that it can be used for longer periods, at closer intervals, and consequently its overall user is better.

With careful attention to the flow of oil and air, there is practically no smoke, so that there is no dirtying of the locomotive and train, nor complaints from those living alongside the railway, as is always the case with steam locomotives. Except for the necessity to turn the locomotive at the terminal stations, the oil fired locomotive has all the advantages of a Diesel locomotive.

All the drivers and firemen, who have driven the oil fired locomotive, appreciate oil firing from the point of view of the engine crew; they stress in particular the definite reduction in physical fatigue for the fireman, the excellent production of steam, which can be rapidly adapted to the needs of the moment, and the cleanliness of the driving cab. As he has no longer to keep firing, the fireman can give more attention to all the auxiliary installations of the locomotive than in the past. But above all, he is able on the oil fired locomotive to help the driver in observing the line and the signals practically without interruption during running. The reduction in physical fatigue also makes it possible to use old, well tried and experienced fireman on the express services, which is very desirable for several reasons.

Finally the oil fired locomotive has another advantage over the Diesel locomotive, which cannot be overestimated. This is that in case of need the oil fired steam locomotive can be reconverted to coal firing at very little cost.

We believe that the D. B. is considering equipping several express series 01¹⁰ locomotives as a first stage with oil firing. It is probable that the reason for this choice is that this series of locomotives is probably the one with the maximum coal consumption. With a monthly mileage of 21 000 km (13 050 miles) and a specific consumption of 15 t/1 000 km run, the 01¹⁰ locomotive consumes a total of 315 t of coal each month. On the other hand a series BR 44 heavy freight locomotive running 10 000 km with a specific consumption of 24 t/1 000 km run, only consumes 240 t per month. Although according to these figures, the absolute coal consumption of a 44 locomotive is about 24 % higher than that of a 01¹⁰ locomotive, the traction department expects from the equipping of heavy freight locomotives with oil firing not only fuel economies, but also special operating advantages for the following reasons:

Experience shows that the coal consumption of a steam locomotive depends to a very large extent on whether running is

normal or not. In the case of the express locomotive, conditions are particularly favourable in this connection. The fireman of an express locomotive, who knows the work to be done by the locomotive on the different sections of the run, can determine precisely how to manage his fire. With good quality coal and an engine in a good state of repair, he has no fears of premature clinking of his fire.

In the case of heavy through freight traffic, the engine crew are faced with much more difficult conditions. On heavily loaded lines, it is not rare for through freight trains to have to make 5 to 10 unexpected stops on a 150 to 200 km (93 to 124 miles) long section. All those concerned will appreciate the unfavourable repercussions of such unexpected stops on the state of the fire grate, even if the steps to be taken in such cases (immediate opening of the auxiliary blower after closing the regulator, etc.) have been observed. After two or three unexpected stops, the fire is generally so dirty up that steam production falls off considerably. The result is that the journey time is exceeded, the running is upset, the staff work overtime and finally there is an appreciable increase in the consumption of coal, all of which consequences can be avoided if it is possible to fit these heavy freight locomotives also with oil firing.

Extensometer measurements, branch of modern metrology,

by Dr. phil. Dipl.-Ing. Herrmann DEUTLER.

(*Die Bundesbahn*, No. 20, October 1956.)(*)

Present state of metrology of strength of materials.

As machinery develops there is always a tendency to attempt to reduce its weight and size while increasing its power. As a result the stresses set up in the various parts of the machine tend to be higher than they used to be. It is not sufficient merely to choose materials of the highest quality, but it is often necessary to be able to check that the performance will be what is expected

by measuring the strength of the complete items.

Today there exists a whole series of processes for measuring the strength of materials, e.g. many problems can be solved by the use of polarised light. This has developed into a complete technique of its own which is called photo-elastic measurement. The technique employs transparent models which are lit up by beams of polarised light and which then throw shadows which appear as « stress bands » rather like the « lines of force » of a magnetic field and which provide information on the loading on the body. Figure 1 shows one of these pictures of a section of rail. It may appear somewhat astonishing that numbers of problems can be solved in this manner with soap bubbles, but this method has been much used since the German Professor L. PRANDTL of Göttingen (1) drew attention to its possibilities. The method consists of placing a film of soap over an opening of given section or height and it is then measured topographically following the contours in the same manner as map makers use for plotting a chain of mountains. Figure 2 shows a soap film carrier, the outline and dimensions of which were calculated for a special case (2). Figure 3 shows a stere-

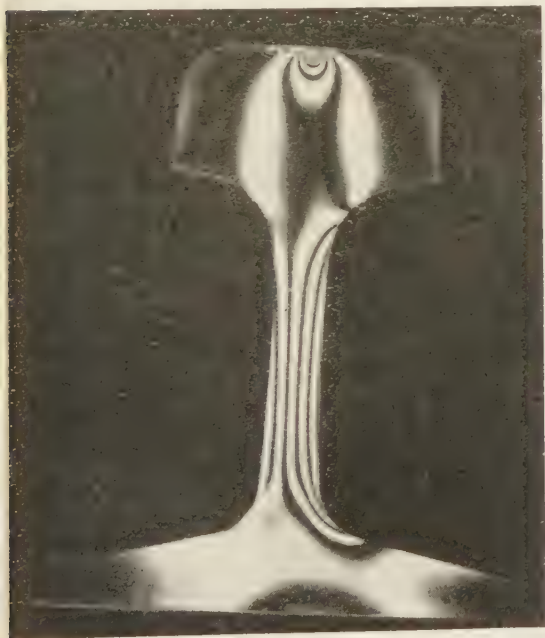


Fig. 1. — Photo-elastic picture of a section of loaded railway rail.

(1) Prof. Ludwig PRANDTL, born 1875 at Freising, died 1953 at Göttingen, conducted original fundamental research into hydro-dynamic and aero-dynamic problems with the aid of polarised light measurements.

(2) Extract from H. DEUTLER, « Ing.-Arch. », 9, (1938), pp. 280-282.

(*) The blocks of the illustrations of this article were very kindly lent by the review « Die Bundesbahn ».

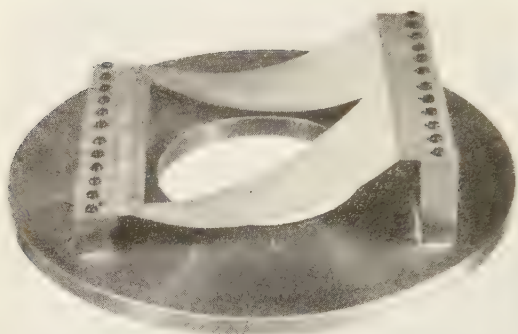


Fig. 2. — Soap film carrier for a triangular section beams subject to torsion.

oscopic photograph of the soap film stretched over this outline. The film has been dusted with lycopodium powder to make it visible and has been lit up by light rays striking it obliquely.

There are other processes for studying strengths of materials such as the use of

20 years, X-rays have also been employed to determine mechanical stresses, and this has extended considerably the techniques formerly available to discover faults in material. All these processes have been mentioned merely to give some idea of the large number of processes that are currently available. To describe them in detail would require many pages, but to keep to the subject of this article it is necessary to deal in detail only with those problems which can be solved by the determination of the *deformation* caused by application of the load.

Tensile measurements. An indispensable part of constructional study.

When a portion of a machine breaks or suffers excessive deformation, it is usually worth while finding out what has caused it, in particular, whether it is the material which is responsible, whether the geometrical design has caused excessive stress, or whether it has been loaded in an incorrect manner. In most cases, fractures leave

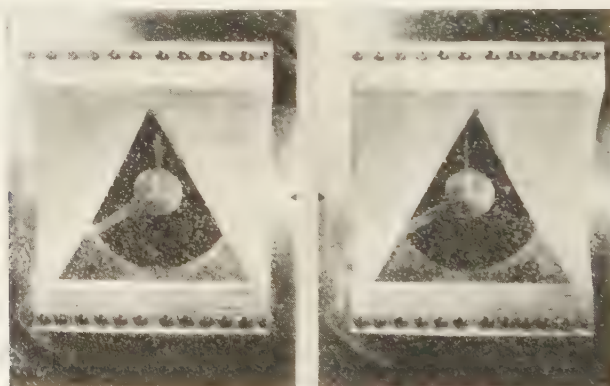


Fig. 3. — Stereoscopic photograph of a soap film for a triangular section.

what is called a « hydro-dynamic analogy ». This consists of comparing the forces induced in a body subjected to torsion with a current of water or an electric field which enables a number of important measurements to be made. For at least

salient marks by means of which, on careful examination, the cause of fracture can be determined. The point of fracture also plays an essential part in the appreciation of failure. If, for example, a machine or a portion of it, always breaks at the same

place, there is *prima facie* evidence that the material is not faulty, but that the geometrical design of the piece is such as to raise the stress at that point enough to cause failure. By studying also the direction of the fracture one can often obtain information on the nature and the direction of the stress causing it. A modification to the shape frequently enables a large number of cases of this kind to be eliminated or at least increases the life of the part so that they become less frequent. But it is also necessary to attempt to deal with this sort of thing preventively, to *avoid* failure ever occurring in the first place, and for this it is necessary to know exactly just what are the stresses which are set up in the body. It is only in this manner that it is possible to predict with any certainty the mechanical behaviour of a piece of equipment.

The word « stress », has a precise sense in the language of mechanics and is used to signify the perpendicular force acting on unit area of the surface of a body. Stresses are, in most cases, spread unequally over parts of a body under load. The distribution can be determined by calculation in the case of standard geometrical shapes subject to simple direct loads. But the more complicated the shape, the more difficult and laborious it becomes to attempt a mathematical solution. In many cases, the constructional technique has no time to wait for mathematical solutions as its problems are not only constantly appearing, but also evolving and changing. It is for this reason that it has been obliged to turn to practical tests in order to obtain information within a reasonable time.

This is done by measurements. To decide what form the measurements should take it is necessary to determine first of all whether the piece to be tested should be the finished article or a model. If the finished article is chosen, the simplest, quickest, and surest method of finding the answers is to proceed to make *extensometer measurements* at predetermined parts on the surface. This makes use of the well-known fact that solid bodies are deformed according to definite laws when subjected to

external loads. Using these deformations as a starting point, conclusions can be drawn as to the internal stresses which are set up.

Attention must be drawn, however, to one particular point. In a process of this nature, one can only measure *surface* extension while any extension that takes place internally cannot be detected thus the measurements will only provide an answer as to the stresses set up on the surface, but, luckily, this is of little disadvantage, since both theory and experience have shown that the stresses responsible for eventual failure, except in a few well-known and defined cases, almost always occur at the surface.

The relation between stress and elongation.

When a three-dimensional body is subjected to external forces, the stresses set up in the body differ in general from one point to another. The theoretician uses a more precise expression. When he considers the stresses at a point he talks of « a state of stresses », and defines it by a mathematical expression which is called a « tensor ». In the same way, the deformation at a given point is also defined by a « tensor ». To elucidate at this point the notion of a « tensor » would be to digress too far, and there would be a danger of losing sight of the main object of this paper in a welter of mathematics. It will be sufficient to accept these definitions as they are and to note that the tensor of stress and the tensor of deformation can be related to one another. This relationship explains the peculiarities of a given material and forms, in effect, a « law » for a particular case. Finding the laws that are applicable is essentially the task of the physicist — verifying their validity is that of the specialist in testing materials.

The best known and most important of the laws on this matter is Hooke's Law ⁽³⁾, which starts from the hypothesis that stress

(3) Robert HOOKE, 1635-1703, the renowned English physicist.

and deformation vary in the same proportions. In the case of steel, which is the most-used material in mechanical construction, the law holds good until the neighbourhood of the « elastic limit » is reached. Just before this limit is reached, molecular displacement takes place which nullifies the validity of Hooke's Law. Current research is seeking to provide an answer as to what is to replace it in these circumstances. These cases, known as « plastic deformation »

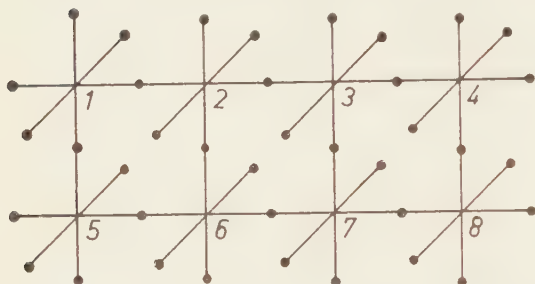


Fig. 4. — Example of a measurement plan.

can take place even under comparatively small external forces, particularly with heated materials. The importance of being able to calculate what plastic deformation will occur in given circumstances in metals is well-known to metallurgists occupied in forging activities. The large quantities of energy which are required for such operations make it well worth while, merely from the economic viewpoint, to be able to calculate in advance what is required. Unfortunately, the mathematical expressions corresponding to these laws are very difficult to establish and it has not been possible up to the present to arrive at a satisfactory solution.

These considerations do not apply in the case of those materials which are popularly known as « breakables ». The latter have no marked elastic limit, but neither do they follow Hooke's Law. All the same, there is a law which covers the behaviour of these types of material ⁽⁴⁾.

(4) H. SCHLECHTWEIG, « Z. angew. Math. u. Mech. », 14 (1934), pp. 1-12.

but it is rarely used, no doubt because its mathematical form is not easy to apply. However, there is an extension and a generalisation of Hooke's Law which can be employed for cast iron, marble, concrete and materials of a similar nature almost up to breaking point. It has enabled, up to the present, problems concerning the bending of members made of these materials, to be solved ⁽⁵⁾, as also the cases of rotating discs ⁽⁶⁾, and cylindrical tubes subjected over all radial compression ⁽⁷⁾. This law also enables the problems of stress in concrete sleepers to be appreciated even though they are somewhat complex.

Returning to the more straightforward problems concerning Hooke's Law, it is worth considering for a moment the conditions which are necessary to enable reliable measurements of the stress distribution to be made.

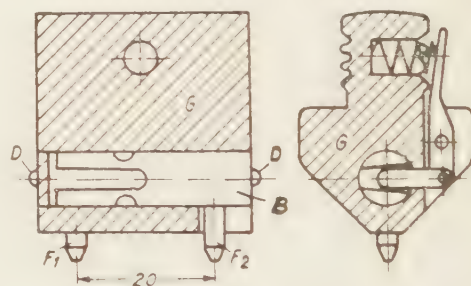


Fig. 5. — Schematic diagram of the operation of the Schwaigerer type mechanical extensometer used at Stuttgart materials testing laboratory.

Practical details.

The practical execution of measurement of elongation requires the establishment of basic points of reference in the immediate vicinity of the required operation. Theoretically speaking, the state of elongation or stress, can only be determined exactly by

(5) H. DEUTLER, « Ing.-Archiv. », 5 (1934), p. 394.

(6) H. SCHLECHTWEIG, « Ing.-Arch. », 5 (1934), p. 7; 6 (1935), p. 365.

(7) E. GLATZEL & H. SCHLECHTWEIG, « Z. angew. Math. u. Mech. », 34 (1954), p. 81.

the use of « infinitely small » bases of measurement. This condition is not, of course, realisable in practice, but it is taken note of by using as short a base as possible, knowing that the shorter the base, the more accurate the measurement. Nevertheless, the attempt to shorten the base must not, for practical reasons, be carried



Fig. 6. — Schwaigerer type mechanical extensometer in use.

too far, as this results in excessively sensitive measuring apparatus being required. The choice of the base length is, therefore, a function of a number of considerations. For example : the probable irregularity of the stress distribution and also the practical limits of the measuring apparatus which is available. As a rule, measurement bases of about 20 mm or slightly less are used.

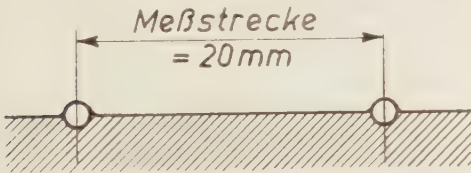


Fig. 7. — Measurement base marked by the imprint made by steel balls.

A particularity of the stress tensor is that it is not sufficient to measure elongation at a particular point in one direction only. To see the whole picture, it is necessary to make measurements in at least three different directions. The most convenient way is to prepare in advance a working

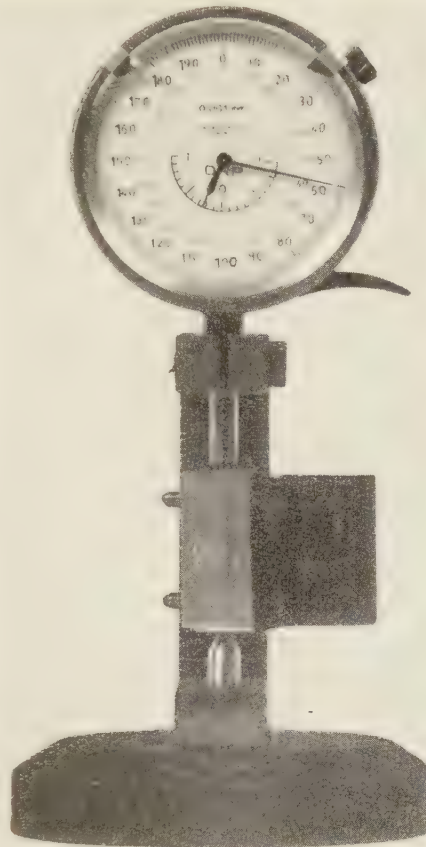


Fig. 8. — Mechanical extensometer being « read off » on a comparometer.

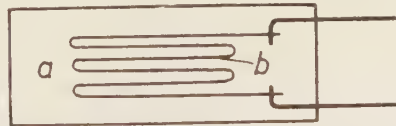


Fig. 9. — Extensometric strain gauges :
(a) resistance wire;
(b) plastic envelope.

plan which is transferred subsequently to the test piece. This simplifies the investigation a great deal.

Figure 4 shows an example of a measure-

TABLE I

Type	Name	Length of base	Amplification
Mechanical types	Huggenberger	10 or 20 mm at choice	1 : 1 200
	Abramson Microcator (Johannson extensometer)	3-50 mm according to model	1 : 1 000 or 1 : 1 500
	Pfender extensometer	variable from 30 to 100 mm	1 : 500
	Schwaigerer extensometer DVL extensometer	20 mm 200 mm	1 : 1 000 without
Pneumatic type	Leiris extensometer	2 mm	1 : 200 000
Optical types	Marten's mirror type	20 to 100 mm according to model	1 : 500
	Junker's Motor Works mirror extensometer	1.5 mm	1 : 10 000
	Berg's mirror extensometer	5 to 20 mm	1 : 10 000
	Freise's optical extensometer	3 and 20 mm	1 : 1 250
Electrical types	Ratzke (DVL) induction extensometer . .	10 mm	1 : 10 000 or 1 : 100 000
	Darmstadt Material Testing Laboratory's extensometer (after Thum, Svenson & Weiss)	up to 0.5 mm	1 : 300 000
	Lehr's photo-electric extensometer . . .	2 mm	up to 1 : 500 000
	Strain gauges (electr. resistance type) . .	variable up to 20 mm	perfect measurements up to 0.001 %
	Acoustic extensometer (Schäfer's system — Maihak & Co., Hamburg)	20 mm	—

ment scheme with eight points of measurement close to one another with their corresponding bases.

Extensometers.

All extensometers work on the principle of measuring the variation in the length

of the base when the test piece is subjected to a load. The variation in the length being extremely small, often attaining only a fraction of a thousandth of a millimetre, an amplifying device is required in order to be able to read the value of the elongation on a scale. The ingenuity of inventors

has resulted in the production of mechanical, optical, mechano-optical, pneumatic, electric, and photo-electric methods of amplification. Naturally, the smaller the measurement base, the greater the degree of amplification required.

Where the load is applied gradually (static loads), the observation arrangement is normally a needle moving round a quadrant as there is sufficient time to make proper observations. In the case, however, of rapidly alternating loads (i.e. dynamic loads), it is necessary to register the elongation automatically and as a function of time. Apparatus for measuring dynamic elongations generally consists of two parts. These are a detector which registers the actual elongation, and a receiver, which registers the signal emitted by the detector after suitable amplification. In the case of rapidly alternating loads, it must not



Fig. 10. — Wheel centre with strain gauges attached.



Fig. 11. — Strain gauges covered with protective mastic on a wheel centre.

be forgotten that unless the registering apparatus has a very small mass, errors due to the inertia of the apparatus will be introduced. This condition favours the provision of a two-part measuring apparatus. Naturally, it is also possible to build static extensometers as two-part instruments and, in fact, it is physically necessary to do so if electrical amplification is used.

It is outside the scope of this article to describe all the measuring instruments which are available and to explain their methods of functioning, but to give the reader a general idea of the apparatus in current use, the essential characteristics of typical instruments have been grouped in Table I. The illustrations also include two extensometers and these have been chosen because of the extreme simplicity in operation. Figure 5 shows the schematic

arrangement of a mechanical extensometer of the Schwaigerer type⁽⁸⁾ which belongs to the Materials Testing Laboratory of Stuttgart. Figure 6 shows the instrument in use. With this type of mechanical extenso-

before and after application of the load. The difference in the readings being the variation in the length caused by the load.

Another very simple form of apparatus is a two-part measuring device. The



Fig. 12. Complete installation at start of measurements.

meter the measurement base is first (fig. 7) marked by the impression made by two $1/16''$ dia. steel balls by means of snap indenter.

The apparatus itself registers by means of hollow bored measuring probes the distance between the points of reference. Its housing (G) carries one of the probes (F_1) while the other probe (F_2) is fixed to an adjustable stud (B) in the housing (fig. 5). Once the impression of the measuring base has been made, the measuring probes are adjusted to this pitch and locked by a special device. The final position of the pitch balls (D) which are fitted to the housing and to the stud respectively is measured by a comparometer graduated in 0.001 mm (fig. 8). A reading is taken

detector is a wire « strain-gauge », the variation in the length of which is apparent in a variation of its electrical resistance, the latter being measured by means of a bridge. Strain gauges are made in the form of an

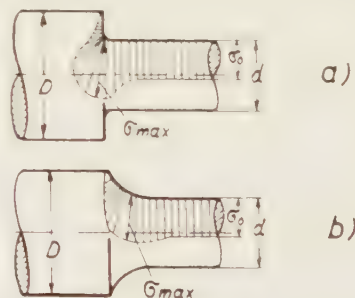


Fig. 13. — Longitudinal stresses at the change of section of a round bar subjected to flexion :

- (a) with a sharp change of section;
- (b) with a gradual change of section.

⁽⁸⁾ S. SCHWAIGERER, « Arch. f. Metallkunde », 3 (1949), pp. 307-308.

envelope of thin plastic material in which is embedded a piece of resistance wire as shown in figure 9. The strain gauge is stuck to the surface of the test piece, the wires follow the surface elongation so that the resistance varies in the same proportion. The accuracy and sensitivity of this apparatus is very high. Changes in length of 0.001 % can be detected and measured with ease.

These two instruments in addition to a number of others, are in general use on the German Federal Railway where they have given excellent results. Figures 10, 11 and 12 show practical examples of the use of strain-gauges. The illustrations show the manner in which the gauges are fixed to measure the stresses set up in a wheel centre by the shrinking-on of the tyre. Figure 10 shows the strain-gauges stuck to one face of the wheel centre. Figure 11 shows the gauges after protection against humidity by means of a special mastic coating; the wires leading to the measuring bridge are shown soldered into place. Lastly, figure 12 shows the wheel centre with all connections made, as it is being inserted into the heated tyre. In the foreground, the connecting bundle and the resistance boxes can be seen and on the extreme left can be seen the bridge.

The importance of stress measurements to the designer.

In the course of the present article attention has been drawn to the usefulness of stress measurements to a rational study

of an element of a mechanism. A typical case is the stress distribution on the surface of a shaft where a change in section occurs when it is subjected to a bending load. Figure 13 shows the difference in the stresses resulting from a given load when the change in section is made gradually or abruptly. Symbol « σ_o » represents the stress caused in the cylindrical portion of dia. « d » while σ_{max} » shows the maximum stress induced at the change of section. It is well-known that the increase of stress caused by changes of section can produce fatigue fractures and the task of the stress analyst therefore, is to develop shapes which produce the smallest possible or even no stress concentrations. In railway construction, there is a large number of working parts of this category, for example, all axles present changes of section of this type. Recently published material on the subject of strength of materials contains hundreds of examples of right and wrong design shapes resulting from interpretations put on to stress measurements.

Summary.

Stress measurement is an important aid to scientific knowledge. It gives a designer a precise basis to work on and the inspector useful information. The progress in the last 25 years has contributed a great deal to diminishing the number of breakages and other damage in machines. It has seemed worthwhile, therefore, to draw attention to this branch of metrology.

Test of a « pendulum » carriage.

Test made by the S.N.C.F. to be able to run at higher speeds, through curves without diminishing the comfort of the travellers or setting up excessive stresses in the track.

Note supplied by the General Management of the French National Railways (S.N.C.F.).

The object of this note is to describe the carriage known as the pendulum carriage which recently has been built by the "Quatre Mares et Buddicom" Works

When a carriage runs at high speed over a curve it tends to be driven outwards by the centrifugal force. Theoretically, this force can be balanced by sufficient super-

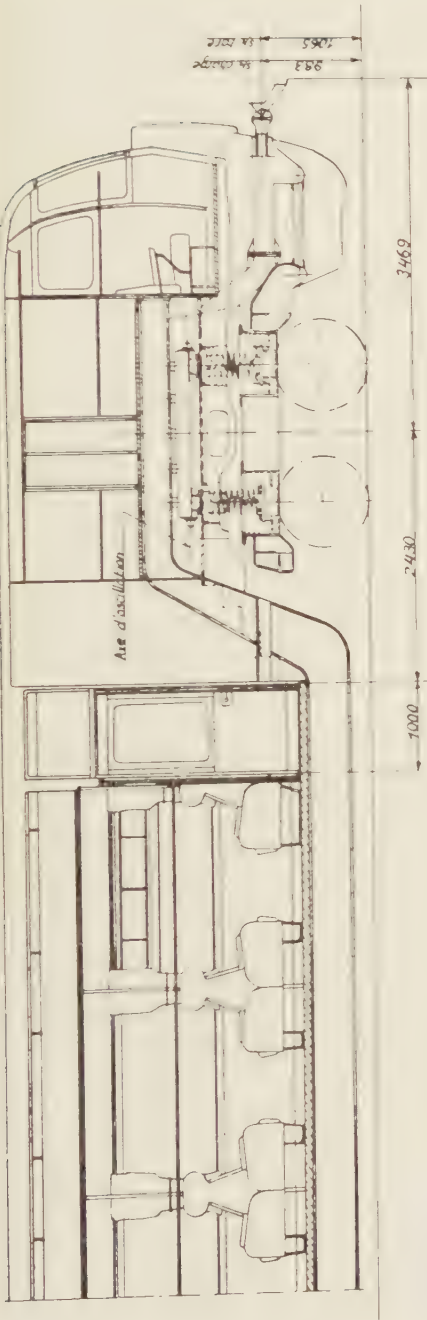


French National Railways coach with pendulum suspension.

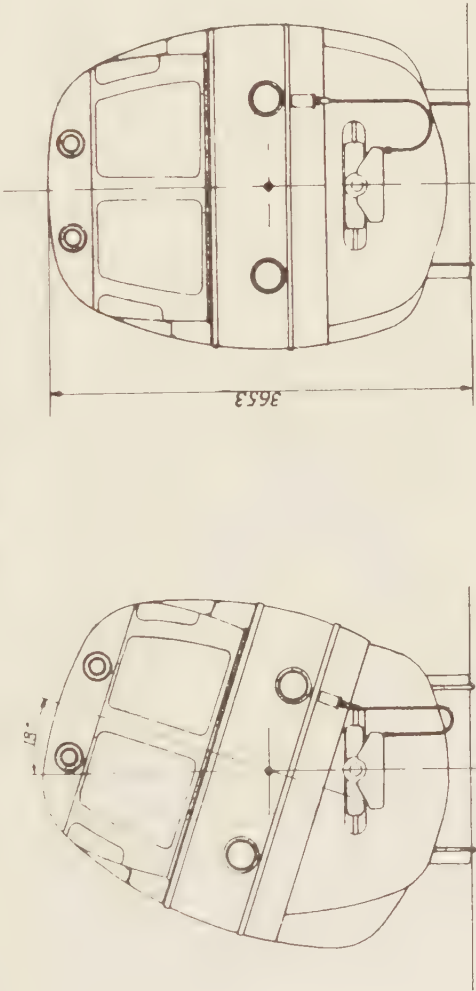
of the Western Region to drawings prepared by the Rolling Stock & Traction Department.

The carriage is based on the following principle :

elevation of the track but this is limited by the amount allowable in practice. It must be remembered that the goods trains run at slow speed and that trains do stop on curves. A marked superelevation

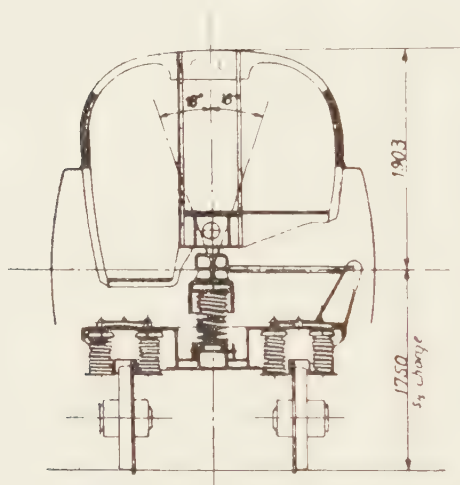


Longitudinal section.



End views.

would have the defect of overloading the inner rail. In practice, a superelevation of as much as 150 mm is allowed for fast lines which corresponds to about 0.10 g unbalanced acceleration. This is a maximum. As an example between Paris and Lyons the speed of 140 km (87 miles)/h is permitted over a curve of 950 m radius with a cant or superelevation of about 14 cm which is equivalent to about 0.07 g not compensated.



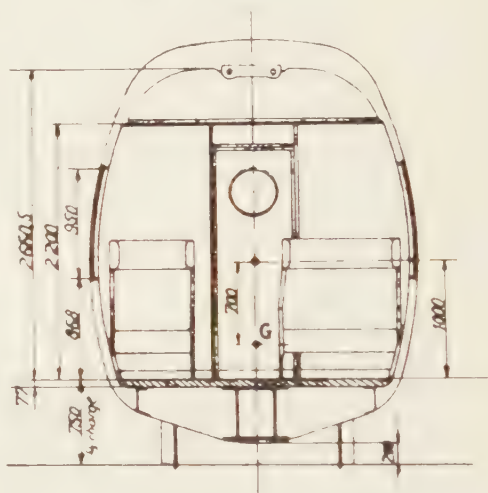
Cross-section on the bogie.

The passenger has to put up with the lateral centrifugal acceleration. Trials made by the S.N.C.F. in 1947 and 1948 have shown that this acceleration was definitely unpleasant for the passenger if it exceeded 0.10 g .

If the speeds on curves are to be increased and for example if curves of 800 m radius with 14 cm cant are to be taken at more than 140 km/h and even speeds of 160 km (100 miles)/h and 180 km (112 miles)/h are to be run over such distances as Paris-Lyons or Paris-

Bordeaux without having to slow down over the curves it becomes necessary to be able to prevent the passenger suffering this centrifugal acceleration.

The problem is the same on the cross country lines with severe curves if the



Cross-section in the body.

present speeds are to be increased. One solution is to make the body able to oscillate like a pendulum about a longitudinal axis located above its centre of gravity. An experiment made in February 1947 between Paris and Etampes at 160 km on curve of 800 m with an oscillating compartment inside a carriage showed that the lateral accelerations of the compartment were in fact considerably reduced. When the carriage itself gave rise to oscillations of 0.24 g , very unpleasant too, the compartment gave oscillations of 0.06 g , i.e. four times less and more than acceptable. The mobile compartment took exactly the inclination of the resultant of the gravity and the centrifugal force.

In view of these encouraging results, it was decided to make a complete test carriage. The Americans in 1942 made tests with a vehicle called "Preco-pendulum" similar in principle, but in which the inclination of the carriage took place by playing on the flexibility of the springs and not by a true pendulum suspension.

To solve the problem completely, at one and the same time the following questions had to be answered :

— To run at high speed, it is not enough to improve the comfort of the passenger, in addition the vehicle must not subject the track to excessive side thrusts. An endeavour was made in carrying out the test to reduce the lateral dynamic loads by making the four wheels of the bogie independent so as to cut out all hunting. It is hoped to avoid the rail wear by using independent wheels, thanks to the lightness of the bogies.

— In the same order of ideas the design was channelled towards a motor coach in order to avoid traction by locomotive with its higher lateral thrusts than those produced by a railcar. In the first carriage about to be tested, the electric motors were not fitted to the carriage and so the first tests will be made with an electric locomotive; the motors will be replaced by equivalent weights in the railcar so that the vehicle under test will be the same as the completed unit as regards weight conditions.

— Finally, the problem of braking inseparable from that of speed leads to the use of disc brakes as giving a better dissipation of the heat resulting from braking. The low axle load of 4.5 t brings down to a moderate level the energy

dissipated when braking from 200 km (124 miles)/h.

* * *

DESCRIPTION

This is a coach carried on two bogies on which the body can oscillate. The body is built on a substantial steel girder with the two ends turned up like a swan neck. The body rests on the two ends of this girder on raised bearings forming part of the bogies. To meet the speeds expected, provision has been made for the body to be able to incline up to 18° on each side of the vertical. The section of the body has its angles rounded off and is such that the body can oscillate without coming outside the loading gauge. The height of the longitudinal axis about which the coach oscillates is 1.75 m above rail level, and the centre of gravity of the oscillating body has been made 70 cm below the longitudinal axis of rotation.

The total weight of the unit is 37 t. In order not to exceed this weight of which a large part is absorbed by the mechanical portion, arrangements were made to build a very light oscillating body and this was achieved by building in light alloy argonarc welded. The coach as completed for trial carries thirty-two passengers.

The principal dimensions are the following :

- distance between bogie centres . . 16 m 860 (55'1")
- length of body . . 22 m 570 (72'9")
- length overall . . 22 m 900 (73'1")

This coach which has many new features and the design of which required

many unknown problems to be solved is an experimental prototype with which tests will be made as regards the comfort of the passengers and at the same time on the stability of the vehicle.

The information obtained will be utilised when studying each field (comfort, behaviour of the bogies on the rails, damping and shocks). It is a research tool for dealing with problems likely to arise on railways in the future.

In order to give details of what can be done with this unit we may quote the following examples :

— At the present time, the Melun-Montereau line via Héricy has curves of 500 to 600 m radius with a speed over them limited to 100 km (62 miles)/h. The new coach will be tested over this line at 110, 120, etc. km (68,74 miles)/h. In the same order of idea the curves of 800 m radius are prescribed for a maximum speed of 130 km (80 miles)/h : tests will be made over such curves at 140 km (87 miles)/h at least. These examples are given as an indication. There may also be problems in connection with increasing the speeds over the secondary lines.

Sixth Working Session (International) on railway vehicles at Aix-la-Chapelle.

(*Die Bundesbahn*, No. 5, March 1957.)

The Institute of Railway Vehicles and Lifting and Transport Installations of the Aix-la-Chapelle Polytechnique, under the management of Professor Max FINK, Dr.-Engineer, during the working session held last year on the 19th and 20th October 1956, gave the foremost place on its agenda to the problem of reciprocal relations between the wheel and the rail.

In view of the enormous dimensions of the field in question — the Railway Administrations belonging to the International Railway Union alone operate more than 250 000 km of line — phenomena of wheel and rail wear have always constituted a particularly important part of scientific research. Numerous publications have long been appearing in the technical journals specially devoted to this subject.

One particularly disagreeable phenomenon of wear for railway working as a whole is the formation of undulatory wear (fig. 1). This not only leads to increased wear of the metal, but also to disagreeable secondary phenomena, amongst others increased noise (screeching) and vibrations over the whole of the vehicles which are most unpleasant for the passengers and extremely undesirable for many kinds of freight and for the vehicles themselves.

Naturally, all possible steps have been taken to eliminate this undulatory wear and to prevent it being set up. To suppress it, the railways have not hesitated to spend considerable sums of money, for example the running of rectifying trains, which are very costly to buy and operate, which repolish the running surfaces of the rails. Naturally first of all, all possible steps are being taken to discover the causes of undulatory wear, but every time it was thought that they had at last been dis-

covered, some new surprises occurred. It is apparent that a whole series of factors play their part together in the formation of undulatory wear.

Research work has been made very difficult owing to the fact that the construction and working of trial sections on which undulatory wear could be studied in different countries and the influence of va-



Fig. 1. Undulatory wear of a rail.

rious factors thereon, costs a great deal and what is even more important, a great deal of time (years). Often results only become apparent after many years, and then it is found that the wrong course had been followed. Thus, the merit of the silent research work carried out in the great Technical Schools is that special methods have been perfected which give results at less cost and in a much shorter time.

WELL, at the request of the British Transport Commission has carried out investigations into friction and wear, taking into account in particular the relations between the wheel and the rail. Taking into account the numerous results obtained by other British engineers, he has endeavoured to bring out the influence of certain factors, for example atmospheric conditions. He has determined the relations between the variations of the coefficients of fric-



Fig. 2. — Amsler wear machine used at the Institute at Aix-la-Chapelle.

This branch of research work received a considerably impetus when it became apparent that in the use of roller bearings, for example in the motor industry, there was also a tendency in certain cases to set up undulatory wear, and in this case again ways and means of studying the causes thereof were sought in order to prevent undulatory wear from the beginning. This is why the Aix-la-Chapelle work session was of such great interest, and about 300 delegates from Germany and other countries were following up the valuable work.

The Head of the Glasgow Mechanical Engineering Research Laboratory, Mr. F.T. BAR-

tion, which also interest brake specialists, and phenomena of wear. For this purpose, he has made use of a test machine which imitates as faithfully as possible the characteristic peculiarities of the movement of rolling with pseudo-sliding, as it occurs on a railway line, especially in the case of driving axles. The whole of the gear box was shut up in an air conditioning installation which made it possible to reproduce different atmospheric conditions (dry, damp, hot and cold weather).

The second reporter, Professor Hilmi ILERI (Istamboul), when he was foreign professor at Aix-la-Chapelle Polytechnique, carried out trials in Professor Fink's Insti-

tute on wear and the formation of undulations on aluminium alloys, by means of Amsler wear machines (fig. 2). He found during these trials that the test machine was a vibrating system which controlled the number of undulations. On the other hand purely friction rolling tests, without any pseudo-sliding produced undulatory wear imperceptible without the control equipment. The pure rolling friction did not arise in the case of railway vehicles with rigid axles. But it is necessary to be very prudent in applying the results of trials on the bench to practical cases.

The main report of Professor Max FINK on the thesis « Roller running without undulatory wear; the rail without undulatory wear » is the result of painstaking research work which has thrown some light on the results of trials of undulatory wear which at times seem contradictory. It appears that undulatory wear must be attributed to the formation of the products of oxidation between the wheel and the rail, which form a sort of film covering the head of the rail, and to the destruction of this film by the rhythmical oscillations of friction. But oscillations of friction only occur when there is pseudo-sliding such as that which occurs mainly at the driving wheels of traction units, as well as on all ordinary railway vehicle axles due to hunting movements on the track.

As in the case of the British investigations, it was found that in studying the behaviour of the wheel and the rail, the presence of oxygen (air) is necessary for the formation of undulatory wear. During comparative trials in a nitrogenous atmosphere, no wear occurred.

It was demonstrated at the same time that owing to the plastic deformation caused by the rolling of the wheels on the rail, the surface of the latter undergoes strong chemical action and can absorb about 270 times more oxygen than iron which has not been deformed.

The film shown, which demonstrated by means of excellent speeded-up sequences the formation of the oxide film and its progressive destruction (always solely in the

case of pseudo-sliding) as well as the accumulation of the particles of oxide torn off, and consequently the beginning of the formation of undulations, was most convincing. Thanks to the trial methods perfected at the Institute by Professor FINK the formation of the film is clearly seen, especially its adhesive capacity and elongation under the influence of different compositions of the steel. Thus a rail with a relative high nitrogen content gives rise to a film which adheres only weakly to the rail head so that the rail has a greater tendency to undulatory wear. An analysis of a great number of rails without undulatory wear showed that they only had a low nitrogen content. In the same way, rails subject to undulatory wear contain a high proportion of nitrogen.

The essential characteristic of the test method perfected by Professor FINK seems to be that it will be possible, at low cost and within a short period of time (a few hours) to study all the repercussions of the composition of the wheel and the rail on wear and the formation of undulations. In fact, it is not sufficient to study the metal used for the rail, but also that used for the tyre if too rapid destruction of the oxide film is to be avoided. With the trials carried out in practical service, the factor « metal of the tyre » can only be studied at great cost, because in this case all the vehicles running over the trial section must have the tyres of their wheels of the same chemical composition. Here we therefore have very great progress due to research in this field.

Similar reasoning applies to the study of roller bearings. Here again an oxide film is produced through wear. This can be destroyed by pseudo-sliding, which only occurs to a very small degree in the case of roller bearings, giving rise to undulatory wear. If undulatory wear occurs in roller bearings, it is generally revealed by the very loud noise set up.

The great interest shown in the discussion by the representatives of the railways, the steel industry and the roller bearing industry, as well as foreign delegates, shows

the importance attached by all of them to the studies carried out. It is to be hoped that the steps, which hitherto have often appeared contradictory, which they will make it possible to take can be combined successfully with the results expected from the research and led to effective prevention of undulatory wear.

On the following day, Professor-Dr. KOTHER (Cologne - Aix-la-Chapelle) demonstrated his flywheel measuring equipment for the determination of journey times and consumption, as well as for the training of drivers for motor vehicles of all kinds. The essential idea is that any method of transport, whether it is question of a train hauled by a steam, electric or thermal engine, or a road vehicle, lorry, coach, or bus, can be represented by an electric motor fitted with a flywheel. The driving motor is designed like a brake dynamo (Tendelsmaschine), and the torque, and consequently the tractive effort, is measured. The speed of rotation corresponds to the running speed. A count is taken

of up and down gradients either by means of a second driving machine acting on down gradients as « motor » and consequently in traction, and on up gradients as « generator » and consequently braked, or again by the movement from zero in the running and braking scale. According to the data given by Professor KOTHER, the accuracy of the measurements is $\pm 1\%$; it is therefore greater than in the case of runs made with different engine drivers in practical services on the same line. In perfecting the equipment, special care was given to use mass produced components in order to keep within satisfactory price limits.

In the future, Professor KOTHER is also considering the training of drivers, especially during the periodical examinations now under consideration for motor drivers: such examinations could then be carried out indoors with very close approximation to actual road conditions. Professor KOTHER is continuing his studies in order to extend the usefulness of this invaluable device.

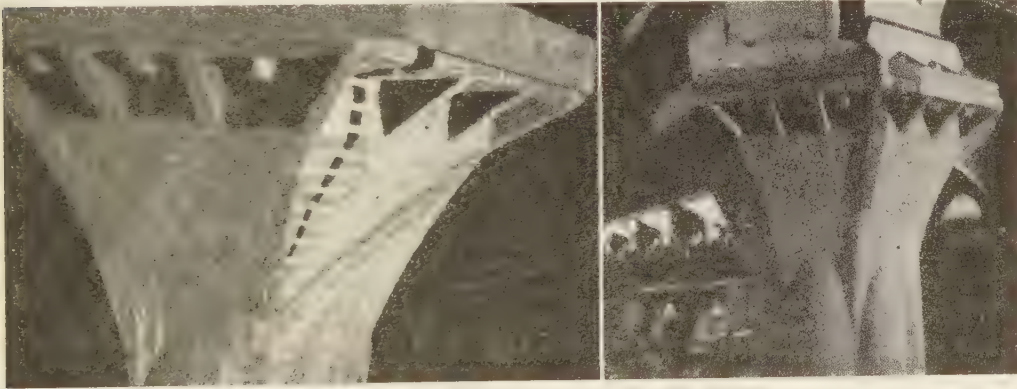
Liverpool Street roof.

Cast-iron columns repaired by Metalock.

(*Modern Transport*, July 13, 1957).

For more than 75 years the roof of Liverpool Street Station, in London, has been supported by cast-iron columns which carry the concentrated load from the arched ribs of the roof. The top three feet or so of each column is shaped like the top half of an inverted cone, moulded and ornamented to conform to the architectural features of the station. During the 1939-45 war, when most of the glass in the roofs

platform 10 had fractured from the top rim to a point about 18 in. down, indicating a tendency of the casting to splay under the load. The top of a column situated on platforms 2 and 3 was also found to be defective but to a greater extent. The bursting action on this column had caused a double fracture in which a V-shaped portion of the column had entirely broken away.



Close-up of the heavily damaged column showing the broken V-piece Metalocked into position ready for final finishing and, right, one of the smaller repairs by Metalock and Metalace processes.

of the London railway stations was removed as a precautionary measure — or by enemy action, this roof withstood the effects of blast from nearby enemy bombing. In a recent routine survey, when some of the cast-iron ornamentation was removed from the columns, it was revealed that four of the columns were cracking at the top.

The columns supporting the roof are tubular; the walls of them are about 1 in. thick and the drainage of the roof is accommodated within the column. A column on

In situ repair.

After deciding that in situ repair by conventional methods was not feasible and that replacing the defective columns would have entailed considerable expense and dislocation of the station working, Metalock (Britain), Limited (whose process was described in *Modern Transport* for August 29, 1953) was consulted with a view to effecting satisfactory in situ repairs, and the work was put in hand on these lines. The necessary

scaffolding and staging were erected by the railway engineering staff.

In the Metalock process, a repair is made by inserting a series of metal « stitches » across the fracture. Along the line of each stitch are drilled rows of holes, one on each side, and with a gun tool a narrow channel is cut between the adjacent holes for each stitch. Into each cavity thus formed a succession of Metalock keys is driven, each key being made from a specially prepared highly ductile alloy of great tensile strength with a temperature coefficient similar to that of the casting, and shaped to fit the cavity exactly.

In the case of the smaller repair at Liverpool Street, a number of keys was driven in by pneumatic hammer one on top of the other in each cavity and these were subsequently peened. Further sheer

strength was added to the repair by using the Metalace process. This consists of drilling a number of holes along the line of fracture, the holes being tapped to receive studs of the same metallurgical characteristics as the keys. As in the case of the keys, the studs were peened and finally the job was finished by a pneumatic grinder.

In the case of the larger repair, the loose piece was clamped into position so that the top of the column was left in the relieved state. This left a wide gap at the fracture, the edges of the broken portion having wasted, but no attempt was made to close the gap as this would have meant inducing new stresses. The repair was made by using longer Metalock keys, the Metalace process being omitted due to the width of the gap. The repairs were completed by two men in eleven days.

Centenary of Crumlin Viaduct.

(*The Engineer*, June 7, 1957.)

We have received from the Western Region of British Railways some details of the Crumlin Viaduct, which has now been in use for a hundred years. The viaduct was constructed on the Taff Vale extension of the Newport, Abergavenny and Hereford Railway, which at that time formed the Newport section of the West Midland Railway. It cost only £41 7 s. per foot run to construct, we are informed, a remarkably low figure even for those days.

The viaduct is in two parts, one of seven spans, the other of three, separated by a short distance less than the length of one of the spans. All the spans are « deck » spans, 150 ft. long, each consisting of four pin-jointed riveted wrought iron Warren trusses. The spans are supported on piers each made up of fourteen 12 in. in diameter cast iron columns braced together. The total length including masonry abutments is 1658 ft.; the greatest height from the bottom of the foundations to the top of the handrail is 208 ft.

Thomas William Kennard was the engineer, and his design, made in competition with several other tenderers, was chosen by Charles Liddell, engineer of the Taff Vale extension. Most of the piers are founded on rock, but two of the highest are on masonry built up from hard gravel, some 14 ft. below ground level.

The iron castings, of which there were 1196 tons in the piers, were made at Kennard's works at Falkirk, Scotland, and the machining and fitting was done at the site of the bridge. There are over 880 tons of wrought iron in the girders and bracings. This wrought iron, having an ultimate tensile breaking stress of between 29 and 30 tons per square inch, came from the Blaenavon Iron Company. The girders were fabricated in the Crumlin Viaduct Works, specially constructed near the site and engaged subsequently in making girder bridges for Italy, Spain, Portugal, and India.

The girders for Crumlin Viaduct were assembled on the ground and lifted singly — a « lift » of 25 tons including stiffening gear — by rope and tackle, at the centre of the piers and subsequently traversed sideways and lowered on to their bearings. The length/width ratio of the compression chord of one of the girders was 200; and although the first girder was successfully lifted to the top of the piers on December 3, 1854, and remained in position without lateral support for several days, the second girder collapsed whilst being hoisted and caused the death of a man riding it. Subsequently banjo bracing was applied to the top chords to stiffen the girders during erection. Twenty men were employed in lifting the girders; it occupied them all one day to lift a girder in the highest part of the viaduct and it was usually set or finally fixed the same night, the men's time costing about £5. The last girder was erected on December 17, 1855.

Each pair of girders was braced laterally in the planes of the top and bottom chords and a decking 26 ft. wide of 6 in. Baltic timbers laid across the tops of the girders carried the two 4 ft. 8 1/2 in. gauge tracks. The viaduct was tested on May 7, 1857, when a maximum central deflection of 1 1/4 in. was recorded under a span loaded with trains on each track totalling some 380 tons. The official opening took place on Whit Monday, June 1, 1857, when, according to an eye-witness's account, « twenty thousand persons were present throughout the day, and cannon were placed on both sides of the valley from which volleys were fired with but slight intermission throughout the day, and caused the mountains to reverberate with their thunder ». The viaduct, railway station and the trains themselves were gaily decorated with bunting and evergreens.

The timber decking was replaced by cross



Crumlin viaduct, which is about 1 1/2 miles west of Pontypool, is in two parts, one of seven spans and one of three spans, all spans being 150 ft. in length. The viaduct was opened on Whit Monday, 1857, amid great rejoicing.

girders, rail bearers and a plate floor in 1868, when also some strengthening of the inner girders was carried out. In 1928 the decking was again renewed and in the following year the line across the viaduct was altered to a single cross-sleeper ballasted track in the middle of the spans. Apart from these and other minor repairs,

the Crumlin Viaduct remain to-day much as it was a century ago. Passage of some of the heavier classes of locomotive is prohibited and the maximum speed of trains is limited to 8 m.p.h. The bridge continues, however, to carry the passengers and mineral traffic of a busy and important railway line.

NEW BOOKS AND PUBLICATIONS.

[385 .02]

DIRECTORY OF RAILWAY OFFICIALS AND YEAR BOOK, 1957-1958. — London : Tothill Press Limited, 33, Tothill Street, Westminster. S.W.1. — One volume of 534 pages (8 1/2 × 5 1/2 inches). Price : 40 s. net.

The Directory of Railway Officials and Year Book, 1957-1958, reaches with the present volume its 63rd year of publication.

The first edition was compiled from official sources in 1895. The present edition of 534 pages covers the text including a general index and the personal index to railway officials, a noteworthy feature of the volume.

This edition should prove as valuable as its predecessors. There has been no marked changes in the layout and general arrangement in this year's edition.

The alteration made last year in the

arrangement of all countries of the world (other than Great Britain and Ireland) in alphabetical order, irrespective of their geographical or political grouping is retained in the present edition. Moreover, the railways in any one country are brought together, whether they are large or small. This should facilitate ready reference.

More details which have become available concerning such countries as Brazil, Egypt, Yugoslavia, Luxemburg and Portugal have been added.

This very interesting publication will be greatly appreciated by all those concerned with railway matters.



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